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Inventory of Direct and Indirect GHG Emissions from Stationary Air Conditioning and Refrigeration Sources, with Special Emphasis on Retail Food Refrigeration and Unitary Air Conditioning

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Nomenclature

A	surface	m^2
F	radiation transfer coefficient	
h	air enthalpy	kJ/kg
h_i	inner heat exchange coefficient	$W/m^2.K$
h_o	outer heat transfer coefficient	$W/m^2.K$
k	thermal conductivity	W/m.K
nd	number of air exchanges in room per 24 hours	
$Q_{defrost}$	extra heat from defrost	W
Q_{fan}	fan motors	W
$Q_{heat-wires}$	anti-sweat heaters	W
$Q_{infiltration}$	infiltration load	W
$Q_{lighting}$	heat dissipation load	W
Q_{load}	total load	W
$Q_{radiation}$	radiation load	W
Q_{wall}	conduction load	W
T	temperature	°C or K
t	insulation thickness	m
U	global heat exchange coefficient	$W/m^2.K$
V	volumetric air flow rate	m^3/s

Greek symbols

ε	surface emissivity	
ρ	density	kg/m^3
σ	Stefan Boltzmann's constant	

Subscript

air,ent	entrapped air
air,ex	air exchange volume
c	condensation (or cold for COP)
case	display cabinet
CR	cold room
e	evaporation

Abbreviation

COP	coefficient of performance

1 Description of commercial refrigeration and stores

1.1   Store categories using refrigeration equipment

Commercial refrigeration equipment is used in different types of stores, for cooled beverage delivery, and food preservation at medium or low temperature. Refrigerating equipment numbers and technologies differ significantly with store types.

Each type or category of store is characterized by a typical structure defined by the average sales surface area, the number of refrigeration equipment, the length of refrigerated cases. Global numbers are established based on Californian statistical data or ratios taken from overall USA numbers.

In order to define a typical store layout, a field study has been carried out in the state of California over a large number of stores, brands, and sale products. Based on the field survey and technical literature analyses, sixteen categories of stores using refrigerating equipment are identified. A total number of 122 stores have been visited during the survey. Table 1.1 presents these categories as well as well the corresponding visited number. Complete list with brand name is reported in Annex 1.

Note: Carbonated soda fountains and vending machines are refrigerating equipment studied independently. They are used in many different stores.

Table 1. 1 Store categories based on field survey

Type	Number of stores visited and described
Grocery supermarkets	54
Minimarkets	3
Pharmacies	10
Convenient stores	12
Liquor stores	5
Butcheries, Pork-butcheries	4
Fishmonger stores	2
Bakeries and Pastries	1
Small size Gas Stations	14
Large size Gas Stations	4
Hotels	8
Motels	5
Bars and Restaurants	1
Carbonated Soda Fountains	-
Vending machines	-
Total	122

1.1.1 Grocery stores or grocery supermarkets

This category gathers two subcategories: grocery stores established primarily for the retailing of food, and large supermarkets that store products other than food, such as clothing or household items. However, since they present the same sales area dedicated to food retailing, these two families are merged in one category referred to as grocery supermarkets. A total number of 54 groceries have been visited. Main brands are Albertsons, Ralphs, Wholefood, Safeway, Walmart, Target, Costco Wholesale, These stores present an average sales area of 4,400 m².

1.1.2 Minimarkets

The mainly visited brands are Smart &Final, foods co. The sales area varies between 300 m² and 1,000 m².

1.1.3 Convenience stores

A convenience store is a small store or shop often located along busy roads. The main visited brands are seven/eleven and AM/PM stores, as well as local stores. An average sales area of 150 m² resulted from survey data processing (visited convenience stores presented sales area varying between 100 and 300 m²).

1.1.4 Liquor stores

A liquor store is the American and Canadian name for a type of convenience stores, which specializes in the sale of alcoholic beverages in the countries where its consumption is regulated. This category presents an average sales area identical to a convenience store. However, a category is dedicated to liquor stores because survey data processing demonstrated that installed refrigeration equipment and systems differ from those found in usual convenience stores.

1.1.5 Pharmacies

The pharmacy is a retail shop where medicine and other articles are sold. The main visited brands are: Wall green, CVS pharmacy, and Rite aid. The sales area varies from 600 m² to 1,000 m². An average sales area of 800 m² is therefore chosen for this category.

1.1.6 Gas stations

A filling station, fueling station, gas station, service station or petrol station is a facility that sells fuel and lubricants for motor vehicles. Most of the visited gas stations had convenience stores selling food and beverages of different sizes. Therefore, two categories are dedicated to gas stations according to the store size and refrigeration load. A first category includes small gas stations, and another one includes mid-size gas stations and gas stations related to commercial centers (for example, Walmart Gas station). The principal brands present in the survey are: 76, Chevron, Mobile, Exxon, Arco,…

1.1.7 Hotels

Hotels of different sizes have been visited during the survey, starting from 1-storey to 12-storey hotels. The principal brands visited are Best Western, Hilton, Marriott, Crowne Plaza, Holiday Inn,… In order to cover the wide range of hotels, a typical hotel lay–out is defined in terms of

room numbers. The typical room number is estimated based on the US Census numbers for hotels and hotel rooms and is found equal to 100 rooms. For a hotel description, the kitchen description is also taken into account.

1.1.8 Motels

The data processing concerning motels is identical to that presented in the hotel section. It is not appropriate to merge these two categories mainly because of significant differences in their kitchen refrigeration features. The visited motel chains are: America's Best Value Inn, Super Motel and Comfort Inn Sunset,...

1.1.9 Bars and restaurants

Bars and restaurants have refrigerating equipment for food conservation and beverage cooling. During the survey, it was not easy to access this equipment for a technical description. The layout of the hotel, which has a restaurant and a bar, is quite similar in terms of refrigeration equipment, except for the ice dispenser at each floor.

1.1.10 Bakeries

Bakeries primarily produce bread and related products, which are then transported to numerous selling points throughout a region. They normally sell beverages and snacks. An average sales area of 125 m² is estimated based on survey data processing.

1.1.11 Butcheries

Butcheries are stores dedicated to prepare meats and other related goods for sale. Several butcheries have been visited (El Cochinito Meat Market, Economy Meat, Veronica Meat Market, Meat Market Carniceria Latina). This category presented an average area of approximately 125 m².

1.1.12 Fishmonger Stores

A fishmonger who sells fish and seafood. This category presents an average area identical to butcheries.

1.1.13 Vending machines

After the data processing, it was more appropriate to group the vending machines in one category to avoid double counting. Since statistics on vending machines present on the Californian market are available, it is possible to evaluate their contribution in commercial energy consumption and refrigerant emissions.

1.1.14 Carbonated Soda Fountains (CSD Fountains)

Data related to CSD fountains have been processed identically to vending machines data, and a category is dedicated to group them.

1.2   Identification of refrigeration systems

1.2.1 Refrigeration systems

Three main technologies of refrigeration systems are used in stores: stand-alone equipment, condensing units, and centralized systems [LIT96].

Stand-alone or plug-in equipment is often a display case where the refrigeration system is integrated into the cabinet and the condenser heat is rejected to the sales area of the supermarket. The purpose of plug-in equipment is to display ice cream or cold beverages such as beer or soft drinks.

Condensing units are small-size refrigeration equipment with one or two compressors and a condenser installed on the roof or in a small machine room. Condensing units provide refrigeration to a small group of cabinets installed in convenience stores and small supermarkets.

Centralized systems consist of a central refrigeration unit located in a machine room. There are two types of centralized system: direct and indirect systems. In a direct system (DX), racks of compressors in the machine room are connected to the evaporators in the display cases and to the condensers on the roof by long pipes. In an indirect system, the central refrigeration unit cools a heat transfer fluid (HTF) that circulates from the evaporator in the machinery room and the display cases in the sales area. The quest for increased energy efficiency and the phase-out of ozone depleting substances have considerably affected refrigeration system design for supermarkets. The traditional CFC and HCFC refrigerants are replaced today with R-404A, R-134a, etc. A number of technical solutions have been tested:
- low GWP refrigerants such as ammonia, propane, and CO_2
- charge minimization by using indirect systems
- improvement of leak tightness of components
- better servicing and beginning of recovery of refrigerants at end of life of equipment.

Still the current centralized direct system is the dominant technology in the US and globally for supermarkets.

1.2.2 Refrigerated cabinets and rooms

Refrigerating equipment is sorted under 3 cabinet technologies: stand-alone equipment or self contained system (SA), display cases (DC) and walk-in coolers (WI).

Display cases and walk-in cabinets can be connected either to centralized system or to condensing unit depending on the equipment size and on the store category, whereas stand-alone equipment are by definition self-contained refrigerating systems.

For each cabinet technology, different types or designs have been identified based on the survey feedback. Technical characteristics and thermal equation have been issued for each type.

1.3 Survey of current refrigeration cases

A survey of 115 stores has been performed from June to November 2007 in order to collect data on existing store structures and types of refrigeration systems and cabinets. The results of this survey provided an abundance of information and allowed estimates to be made of current electricity consumption for the operation of refrigeration cabinets either as direct consumption by the cabinets (lighting, fans, anti-sweat heaters, defrosting) or by refrigeration compressors and condenser fans in order to provide refrigeration to these cabinets. The survey was performed for both remotely operated cases (DC and WI) and self-contained refrigerated equipment (SA).

1.3.1 Survey contents and data collection

Display cases (DC) and stand-alone equipment (SA) include low temperature single-deck, low temperature multi-deck, low and medium temperature glass door, medium temperature single-deck, medium temperature multi-deck, service cases, and specialty cases. Specifications include the make and model, case length, blown air temperature, saturated suction temperature, and all are included in the database.

The product display has been divided into the following categories: dairy, deli, meat product, beverage, bakery, frozen food, and ice cream. In many instances, a cabinet can be used for several of these products. Where the product displayed affects the operating temperatures or refrigeration loads, a separate entry (in the data base) for the case is provided for each product. If the specified temperatures and refrigeration loads are the same for multiple products, the products used are noted in the description.

Survey data have been collected and regrouped as a function of the refrigeration cabinet type. Hence, surveys are presented separately for display cases, stand-alone equipment, walk-in and storage rooms.

During the survey, store data have been recorded and included store's brand name, location and average sales area. For the presently manufactured refrigeration equipment, several information have been collected as presented thereafter:
- Brand name of the equipment manufacturer
- Equipment model number: *ex:* for a TRUE equipment, GDM–35
- Temperature level (medium, low)
- Equipment position: horizontal, vertical, semi-vertical
- Open or closed type equipment
- For closed type, the number of doors is recorded whereas for open type, the total length of the equipment is estimated.
- Equipment capacity and dimensions: capacity in cf or liters, height, width and length.
- Refrigerant type and charge.
- Product type (dairy, deli, bakery, salads, floral, meat, drinks, ice cream…).

The purpose of remote or self-contained refrigerated display cases in a store is to provide temporary storage for perishable foods prior to sale. Most of the design characteristics and general shape and layout of display cases are based on marketing specifications and constraints. The configuration of display cases falls into essentially four different categories.

- **Tub**: The tub case is often used for the storage and display of frozen foods and meats. Tub cases operate at a very uniform temperature and require the lower refrigeration capacity per foot of any display case type. The primary disadvantage of the tub is a low product storage volume per square foot of sales area.

- **Open-front multi-deck**: This case type possesses the largest storage volume per square foot of floor area, because of the use of an upright cabinet and shelves. Refrigeration capacity required for multi-deck cases is very high, including a large latent load portion due to the entrainment of ambient air in the air curtain passing over the opening of the case.

- **Glass door reach-in**: The reach-in case has glass doors over the opening of the case; these must be opened for product removal and stock. Reach-in cases are used in supermarkets primarily for frozen foods, because of their ability to contain the cold refrigerated air, which reduces the "cold aisle" problem.

- **Single-deck or service**: Open single-deck cases are commonly used for display of fresh meat products. The service display case is a single-deck case equipped with sliding doors in the back for access by serving people and a glass front to show product to customers. Cases of this type are commonly seen in the deli and meat departments of supermarkets.

Display cases have been developed and refined for specific merchandising applications, and cases of each type listed above exist specifically for the storage and display of specific food types.

To allow definition of baseline refrigeration equipment, survey data have been processed based on technical data of leading refrigeration cabinet manufacturers in the United States. Data processing showed capacities and dimensions found in the stores that could be different from data gathered on websites of equipment manufacturers. To take into account these differences, interpolations have been made on the refrigeration capacity as well as the input power.

When an equipment description is identical to a model listed in the table (except for its manufacturer), input power and refrigerant data are directly applied to the studied equipment.

1.3.2 Stand-alone equipment

One objective of the survey is to define baseline stand-alone equipment models depending on description parameters stated above. Leading manufacturers of stand-alone equipments are: True Manufacturing, Beverage Air, and Hussmann Corporation. The stand-alone cases listed have been categorized into 23 models presented in Table 1.2, each model having a number starting from 1 to 23.

1.3.3 Display cases

Leading refrigeration equipment manufacturers of display cases are: Hussmann Corporation, Hill Phoenix, Tyler Refrigeration Corporation, and Kysor Warren. 14 baseline display case models are defined in Table 1.3, each model having a number starting from 1 to 14.

1.3.4 Walk-ins

Selections include storage walk-ins, walk-in boxes with glass doors (e.g., dairy, beverage and floral boxes), preparation areas that may be fully enclosed or have one side open to the sales area, and other perimeter zones that are air conditioned from the refrigeration system (e.g., bakery prep areas, pharmacy, etc.). Specifications include the make and model (for components in the library), size, temperature, location, reach-in doors, walk-in doors, refrigeration load, lighting, evaporator coils, defrost type and control, fans, and internal loads. For walk-in (WI), 5 baseline categories are found and listed in Table 1.4.

Table 1. 2 Baseline stand-alone equipments list

Number	Brand Name	Model	T° level	Open/closed	Position	Door #	Capacity (liters)	Length (m)	Height (m)	Width (m)	Product type	Refrig type	Refrig charge (g)	Cooling Capacity (HP)	Product T(°C)	Evaporation T(°C)
SA_1	(TRUE)	GDM-7	Medium	closed	Vertical	1	200	0.8	1.0	0.6	drinks, salads, deli, dairy	R134A	300	0.2	2.0	-13
SA_2	(TRUE)	GDM-10	Medium	closed	Vertical	1	300	0.8	1.4	0.7	drinks, salads, deli, dairy	R134A	300	0.2	2.0	-13
SA_3	(TRUE)	GDM-23	Medium	closed	Vertical	1	600	0.8	2.0	0.7	drinks, salads, deli, dairy	R134A	300	0.33	2.0	-13
SA_4	(TRUE)	GDM-35	Medium	closed	Vertical	2	1000	1.6	2.0	1.0	drinks, salads, deli, dairy	R134A	300	0.5	2.0	-13
SA_5	(TRUE)	GDM-72	Medium	closed	Vertical	3	2000	2.4	2.0	2.0	drinks, salads, deli, dairy	R134A	550	0.5	2.0	-13
SA_6	(TRUE)	TCGR-50	Medium	closed	S-Vertical	2	800	1.6	1.2	1.3	salads, deli, bakery	R134A	300	0.5	2.0	-13
SA_7	Hussman	SHM	Medium	open	S-Vertical			1.8	1.2	0.6	drinks, salads, deli, dairy	R134A	800	0.5	2.0	-13
SA_8	(TRUE)	TAC-48	Medium	open	Vertical		1000	1.2	2.0	1.2	drinks, salads, deli, dairy	R404A	1000	1	2.0	-13
SA_9	Hussman	DDS8	Medium	open	Vertical			2.4	2.5	2.1	drinks, salads, deli, dairy	R22	1400		2.0	-13
SA_10	(TRUE)	THF-41FL	Low	closed	Horizontal	2	300	1.6	1.0	0.8	frozen	R134A	1200	0.75	-20.0	-35
SA_11	(TRUE)	GDM-23F	Low	closed	Vertical	1	600	0.8	2.0	0.7	frozen	R404A	600	0.75	-20.0	-35
SA_12	(TRUE)	GDM-35F	Low	closed	Vertical	2	1000	1.6	2.0	1.0	frozen	R404A	1000	1	-20.0	-35
SA_13	(TRUE)	GDM-72F	Low	closed	Vertical	3	2000	2.4	2.0	2.0	frozen	R404A	2500	1.5	-20.0	-35
SA_14	Bev Air	FC-45	Medium	closed	Vertical	2	1200	1.6	2.0	0.7	Floral	R134A	600	0.33	2.0	-13
SA_15	Hussman	ISMGG	Medium	open	Horizontal			1.7	1.0	1.0	deli, dairy	R22	2300		2.0	-13
SA_16	Hussman	ISFGG	Low	open	Horizontal			1.7	1.0	1.0	frozen	R22	2300		2.0	-13
SA_17	(TRUE)	GDM41/45	Medium	closed	Vertical	2	1200	1.6	2.0	0.8	drinks, salads, deli, dairy	R134A	500	0.5	2.0	-13
SA_18	(TRUE)	GDM 12	Medium	closed	Vertical	1	340	0.8	1.6	0.7	drinks, salads, deli, dairy	R134A	250	0.2	2.0	-13
SA_19	Bev Air	MT12	Medium	closed	Vertical	1	400	0.8	1.6	0.6	boissons	R134A	300	0.2	2.0	-13
SA_20	Bev Air	UR30	Medium	closed	Vertical	1	700	0.8	0.9	0.7	boissons	R134A	200	0.2	2.0	-13
SA_21	LEER	Model 45	Low	closed	Vertical	2	1200	1.6	2.0	1.2	Ice maker	R134A	300	0.33	-25.0	-35
SA_22	Bev Air	DD68	Medium	closed	Horizontal	2	800	1.6	1.0	0.7	Beer	R134A	300	0.33	2.0	-13
SA_23	(TRUE)	GSM	Medium	closed	Vertical	1	700	0.8	2.0	0.8	Tour Patissier	R134A	300	0.5	2.0	-13

Table 1. 3 Baseline display cases list

Number	Brand Name	Model	T° level	Type	Position	Door #	Length (m)	Height (m)	Width (m)	Cooling Capacity (BTUh/FT)	Cooling Capacity (W/m)	Refrig Charge (Lb/FT)	Refrig Charge (kg/m)	Product type	T evaporation (°C)	Tproduct (°C)
DC-1	TYLER	N6D	Medium	open	Vertical		1.2;1.8;2.4;3.6	2.0	1.0	1270	1222	1	1.55	Dairy/Deli/Produce/Juice	-13	2
DC-2	HUSSMAN	RM	Medium	closed	Vertical	1;2;3	0.8	2.1	1.1	324	312	0.35	0.52	Dairy/Deli/beverages	-13	2
DC-3	HUSSMAN	E2	Medium	open	1/2Vert		1.2;1.8;2.4;3.6	1.2	1.2	945	909	0.23	0.34	Deli/Pizza/Floral/Juice	-13	2
DC-4	HUSSMAN	SMGV	Medium	closed	1/2Vert		1.2;1.8;2.4;3.6	1.3	1.1	210	202	0.19	0.28	Meat/delicatessen	-13	2
DC-5	TYLER	NM/NMG	Medium	open	Horizontal		1.8;2.4;3.6	1.0	1.0	432	416	0.26	0.39	Meat/deli/Produce	-13	2
DC-6	HUSSMAN	F6L	Low	open	Vertical		1.2;1.8;2.4;3.6	2.0	1.2	1755	1688	0.44	0.66		-35	-20
DC-7	HUSSMAN	RLN	Low	closed	Vertical	2;3;4;5	0.8	2.1	2.0	503	484	0.34	0.51	Frozen	-35	-20
DC-8	TYLER	NFNG	Low	closed	Horizontal		2.4;3.6	0.9	1.0	366	352	0.28	0.42	Frozen	-35	-20
DC-9	TYLER	NPW	Medium	open	Horizontal		2.4;3.6	0.8	1.8	776	747	0.51	0.76	Produce	-13	2
DC-10	HUSSMAN	P2X	Medium	open	Vertical		1.2;1.8;2.4;3.6	1.9	1.0	910	875	0.38	0.57	Fruits/Produce	-13	2
DC-11	HUSSMAN	F2XLG	Low	open	1/2Vert		1.2;1.8;2.4;3.6	1.4	1.2	2050	1972	0.44	0.66	Frozen meat/Seafood	-35	-20
DC-12	HUSSMAN	DSFM	Medium	closed	1/2Vert		1;2;3;4;	1.2	1.0	500	481	0.33	0.49	Seafood	-13	2
DC-13	Hussman	FI	Low	open	Horizontal		2.4;3.6	0.9	1.5	480	462	0.34	0.51	Frozen	-35	-20
DC-14	ZeroZone	RMCP30FL	Medium	closed	Vertical	2;3;4;5	0.8	2.1	0.9	1220	1174	0.43	0.64	FLORAL	-13	2

Table 1. 4 Baseline walk-in cases list.

Number	T° level	Type O/C	Type R/C	Door #	Length (m)	Height (m)	Width (m)	Product DH (kg/kJ)	Product mass flow (kg/m3 24h)	Product T(°C)	Tevaporation (°C)
WI-1	Medium	open	0		1	2.5	3	45	20	2	-13
WI-2	Medium	closed	0	12	1	2.5	3	45	20	2	-13
WI-3	Low	closed	0	12	1	2.5	3	55	15	-20	-35
CSR-1	Medium	closed	1		1	3.5	6	45	20	2	-13
CSR-2	low	closed	1		1	3.5	6	55	15	-20	-35

Once the baseline refrigeration equipment, self-contained or remote refrigerated equipment are described, it is possible to draw typical layouts for the 16 store categories defined in Section 1.1. The next section presents the grocery supermarket layout based on equipment listed in Tables 2 to 4.

1.4 Typical grocery store lay-out

A representative grocery supermarket layout is shown in Figure 2. Refrigerated fixtures are located throughout the store, because of the large amount of perishable food products that are sold. These fixtures fall into 3 categories, stand-alone equipment, display cases, and walk-in storage coolers. Stand-alone equipment and display cases are located on the sales floor and are designed to refrigerate food products while providing a place to merchandise them. Walk-in coolers are used to store food products during the time period between receiving the product and placing the product out for sale.
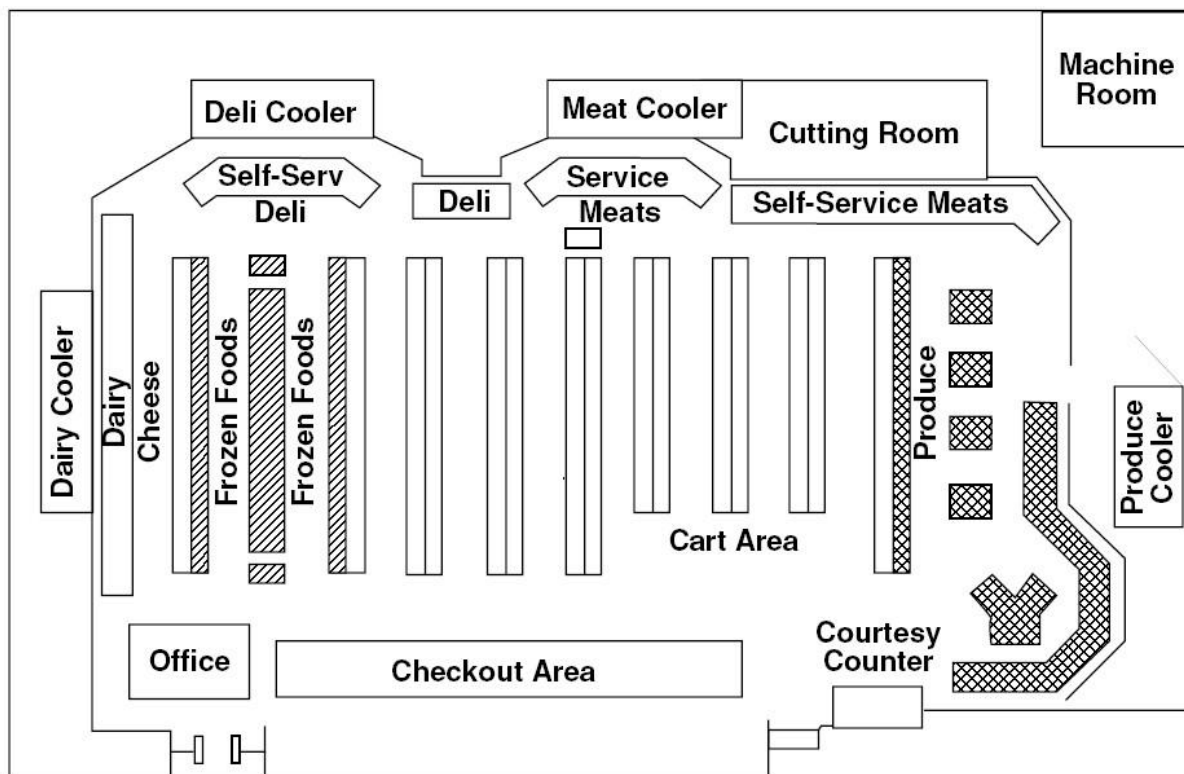
Figure 1. 1 Lay out of the refrigerated fixtures in a supermarket [ORNL04].

A typical arrangement of refrigerating equipment in a grocery is shown in Figure 1.1. Display cases, of a variety of configurations and products, are generally used in the sales area and are located at the periphery of the store near their associated walk-ins. The survey data processing enabled the definition of a typical grocery refrigeration configuration presented in Tables 1.5, 1.6, and 1.7.

Table 1. 5 Self-contained refrigerating equipments found in a grocery store

Stand Alone equipments		
Description	Model	Number of equipment
Medium temperature self contained closed vertical case for drinks salads, deli and dairy with a capacity of 200 liters	SA-1	3
Medium temperature self contained closed vertical case for drinks salads, deli and dairy with a capacity of 300 liters	SA-2	2
Medium temperature self contained closed vertical case for drinks salads, deli and dairy with a capacity of 600 liters	SA-3	1
Medium temperature self contained open vertical case for drinks salads, deli and dairy with a capacity of 1000 liters	SA-8	1
Medium temperature self contained open vertical case for drinks salads, deli and dairy with a capacity of 2000 liters	SA-9	1
Medium temperature self contained open horizontal case for deli and dairy with a capacity of 1500 liters	SA-15	1
Medium temperature self contained closed vertical case for drinks salads, deli and dairy with a capacity of 1200 liters	SA-17	1
Medium temperature self contained closed vertical case for drinks salads, deli and dairy with a capacity of 340 liters	SA-18	1
Self contained Ice maker with a capacity of 1200 liters	SA-21	1

Table 1. 6 Display cases equipments found in a grocery store

Display cases equipments		
Description	Model	Length (m)
Medium temperature Open-front multi-deck vertical display case for dairy, deli, juice and drinks	DC-1	75
Medium temperature Glass door reach-in multi-deck vertical display case for dairy, deli, juice and drinks	DC-2	7
Medium temperature Open-front single-deck semi-vertical display case for deli, pizza floral and juices	DC-3	15
Medium temperature Glass door reach-in single-deck semi-vertical display case for meat and delicatessen	DC-4	20
Medium temperature Open Tub case for meat and delicatessen	DC-5	10
Low temperature Open-front multi-deck vertical display case for frozen products	DC-6	4
Low temperature Glass door reach-in multi-deck vertical display case for frozen products	DC-7	86
Medium temperature Open Tub case for produce	DC-9	17
Medium temperature Open-front multi-deck vertical display case for produce	DC-10	27
Medium temperature Open-front single-deck semi-vertical display case for seafood	DC-12	5
Low temperature Open Tub case for frozen products	DC-13	17
Medium temperature Glass door reach-in multi-deck vertical display case for floral	DC-14	3

Table 1. 7 Walk-in and cold rooms found in a grocery store

Walk In and Storage Rooms		
Description	Model	Length (m)
Medium temperature Open-front multi-deck walk in for dairy, deli, juice and drinks	WI-1	4
Medium temperature Glass door reach-in multi-deck walk in for dairy, deli, juice and drinks	WI-2	12
Low temperature Glass door reach-in multi-deck walk in for dairy, deli, juice and drinks	WI-3	12
Medium temperature Cold Storage room	CR-1	60
Low temperature Cold Storage room	CR-2	18.5

1.5 Typical layout of small stores

Similarly to grocery stores, the survey data processing enabled the definition of a typical refrigeration layout for each of the 15 categories defined previously. These layouts are described in Tables 1.8, 1.9, and 1.10.

Table 1. 8. Self-contained refrigerating equipment distribution for different store categories.

Stand Alone	SA-1	SA-2	SA-3	SA-4	SA-5	SA-6	SA-7	SA-8	SA-9	SA-10	SA-11	SA-12	SA-13	SA-14	SA-15	SA-16	SA-17	SA-18	SA-19	SA-20	SA-21	SA-22	SA-23
Grocery	3	2	1	0	0	0	0	1	1	0	0	0	0	0	1	0	1	1	0	0	1	0	0
Large Supermarket	2	2	0	3	0	0	0	0	0	2	1	0	0	2	0	1	1	0	0	0	2	0	0
Pharmacy	1	1	0	0	0	0	0	0	0	1	0	0	0	0	0	0	0	0	1	0	1	0	0
Convenience	0	0	0	2	0	0	0	0	1	2	0	0	0	0	0	0	0	0	1	0	1	0	0
Liquor Store	0	1	0	1	1	0	0	0	0	3	0	0	0	0	0	0	1	0	1	0	1	0	0
Minimarket	1	2	1	0	0	0	0	1	0	1	1	0	1	0	1	0	0	0	0	0	2	0	0
Small Gas Stat	0	0	1	0	0	0	0	0	0	0	0	0	0	0	0	1	0	0	0	0	0	0	0
Center Gas Stat	1	0	0	1	0	0	0	0	0	1	1	0	0	0	0	0	1	0	1	0	1	0	0
Hotel	2	0	0	1	0	0	0	0	0	0	0	1	0	0	0	0	0	0	0	0	4	1	0
Motel	0	0	0	0	1	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	3	0	0
Butchery	0	0	0	1	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Fishmonger Store	0	0	0	1	0	0	0	0	0	0	0	0	0	0	0	0	0	0	1	0	1	0	0
Bakery	0	0	0	2	1	0	0	0	0	1	0	0	0	0	0	0	0	0	0	0	0	0	1
Restaurants Bar	2	0	0	1	0	0	0	0	0	0	0	1	0	0	0	0	0	0	0	0	1	1	0
Vending Machine	0	0	0	1	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Soda Fountain	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	1	0	0

Table 1. 9. Distribution of refrigerated display cases for different store categories

Display Case	DC-1	DC-2	DC-3	DC-4	DC-5	DC-6	DC-7	DC-8	DC-9	DC-10	DC-11	DC-12	DC-13	DC-14
Grocery	75	7	15	20	10	4	86	0	17	27	0	5	17	3
Large Supermarket	0	10	8	0	6	0	11	0	0	1	0	0	1	0
Pharmacy	0	1	0	0	0	0	3	0	0	0	0	0	0	0
Convenience	1	0	0	2	0	0	1	0	0	3	0	0	0	0
Liquor Store	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Minimarket	4	0	0	0	0	0	3	0	0	2	0	0	2	0
Small Gas Stat	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Center Gas Stat	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Hotel	0	0	0	3	0	0	0	0	0	0	0	0	0	0
Motel	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Butchery	0	0	2.5	2.5	0	0	0	0	0	0	0	0	0	0
Fishmonger Store	0	0	5	0	0	0	0	0	0	0	0	0	0	0
Bakery	0	0	0	5	0	0	0	0	0	0	0	0	0	0
Restaurants Bar	0	0	0	3	0	0	0	0	0	0	0	0	0	0
Vending Machine	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Soda Fountain	0	0	0	0	0	0	0	0	0	0	0	0	0	0

Table 1. 10 Distribution of walk-in and cold storage rooms for different store categories.

Walk IN	WI-1	WI-2	WI-3	CR1	CR2	Sales Area (m²)
Grocery	4	12	12	60	19	2500
Large Supermarket	0	6	9	27.5	9	8500
Pharmacy	3	8	4	0	0	800
Convenience	0	9	2	0	0	150
Liquor Store	0	14	0	0	0	153
Minimarket	0	14	21	27.5	9	1145
Small Gas Stat	0	2	0	0	0	25
Center Gas Stat	0	7	2	0	0	100
Hotel	0	0	0	6	3	
Motel	0	0	0	0	0	
Butchery	0	0	0	3	0	125
Fishmonger Store	0	0	0	3	0	125
Bakery	0	0	0	0	0	125
Restaurants Bar	0	0	0	6	3	
Vending Machine	0	0	0	0	0	
Soda Fountain	0	0	0	0	0	

2 Method for Energy Consumption Calculation

2.1 Introduction

Supermarkets represent one of the largest energy-intensive building groups in the commercial sector, consuming 2 to 3 million kWh annually per store [BAX03]. Several studies have shown that annual electricity consumption ranges from 1 to 1.5 million kWh per store for refrigeration [LIT96]. A typical electricity usage of a grocery in the U.S. shows that 39% is used for refrigeration, 23% for lighting, 11% for cooling, 4% for ventilation, 13% for heating, and 10% for miscellaneous applications (cooking, water heating, …) as shown in Figure 2.1.
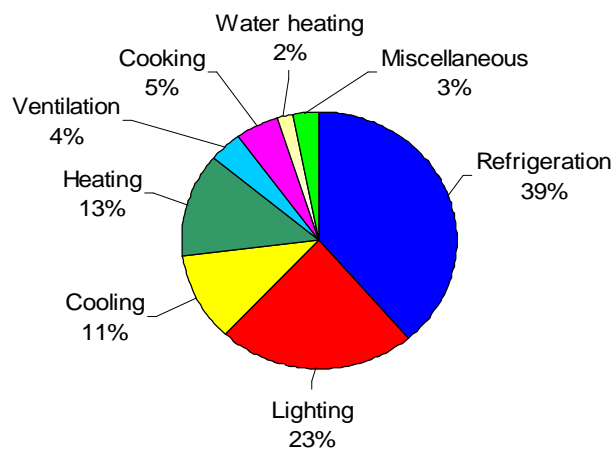

Figure 2. 1 Typical electrical energy usage in a grocery store in USA [LIT96].

Recent field tests tend to confirm that this figure is still a good estimate. Data from a field test in a 50,000 ft²-store in Southern California indicate annual usage of about 1,500,000 kWh for all refrigeration including case lights, fans, heaters, etc [ORL04].
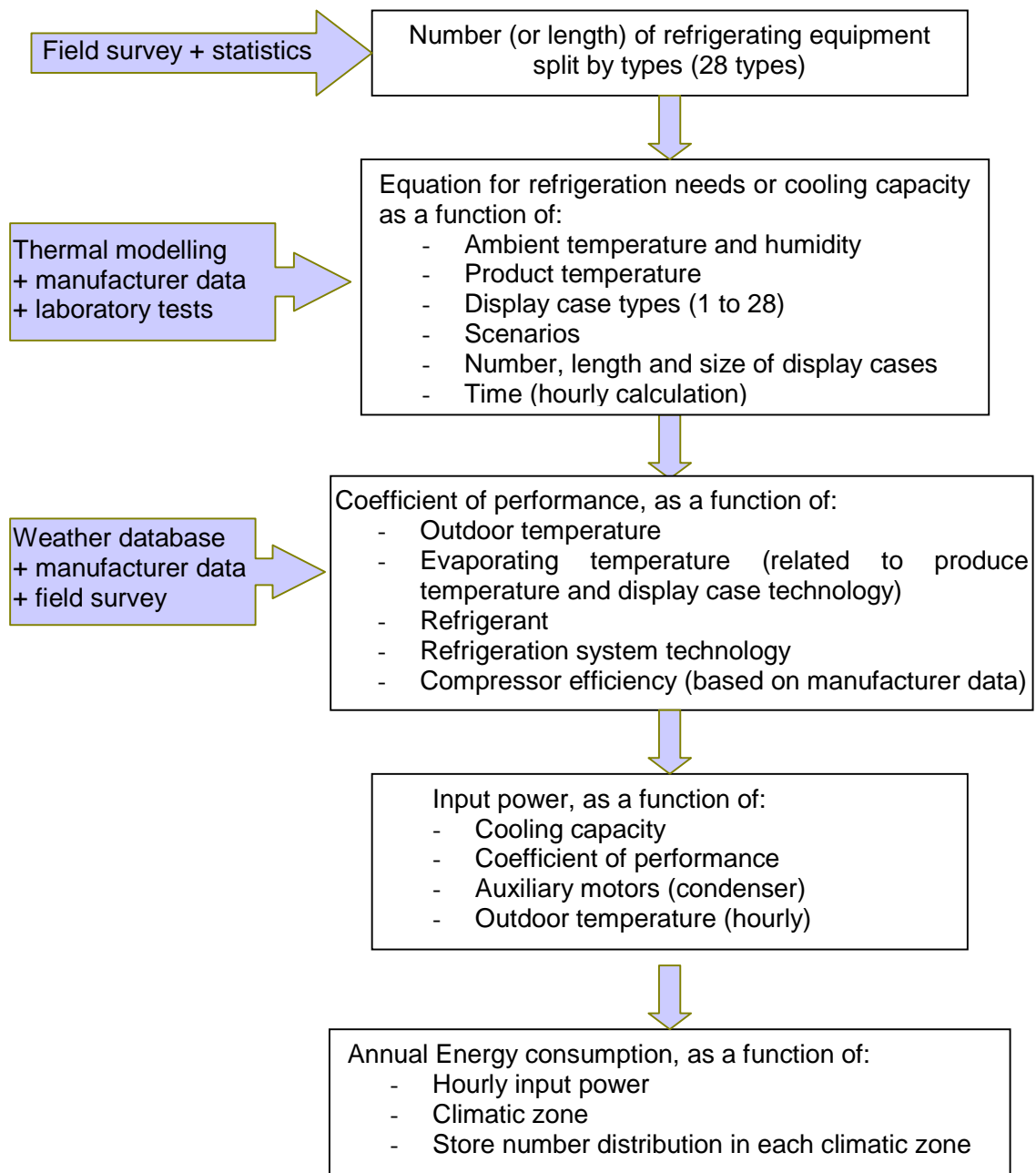
The approach for energy consumption calculation in commercial refrigeration, detailed in this report, is qualified as "bottom – up approach". In order to simulate energy efficiency improvement of refrigeration equipment, each element in the energy consumption chain has to be considered and described in detail.

The energy consumption calculation is based on the evaluation of refrigeration loads, hour by hour, on a given year, taking into account weather conditions (temperature and humidity) of the 8 California climatic zones.

Each type of store (16 families) have been calculated independently, when the layout of refrigeration equipment in each store has been issued.

2.2   Energy consumption calculation

The method for energy consumption calculation is illustrated by the following algorithm. The method is applied for each type of store.

Field survey + statistics →

Number (or length) of refrigerating equipment split by types (28 types)

Thermal modelling + manufacturer data + laboratory tests →

Equation for refrigeration needs or cooling capacity as a function of:
- Ambient temperature and humidity
- Product temperature
- Display case types (1 to 28)
- Scenarios
- Number, length and size of display cases
- Time (hourly calculation)

Weather database + manufacturer data + field survey →

Coefficient of performance, as a function of:
- Outdoor temperature
- Evaporating temperature (related to produce temperature and display case technology)
- Refrigerant
- Refrigeration system technology
- Compressor efficiency (based on manufacturer data)

Input power, as a function of:
- Cooling capacity
- Coefficient of performance
- Auxiliary motors (condenser)
- Outdoor temperature (hourly)

Annual Energy consumption, as a function of:
- Hourly input power
- Climatic zone
- Store number distribution in each climatic zone

The cooling capacity of display cases is provided by a large vapor compression refrigeration system. The operating characteristics and energy requirements of the refrigeration system are directly related to the refrigeration capacity necessary to maintain display case temperature.

There are two principal temperature levels in supermarkets: medium temperature for preservation of chilled food and low temperature for frozen products. Chilled food is maintained between 1°C and 14°C, while frozen food is kept between -12°C to -18°C, depending on the country. The evaporation temperature, for a medium-temperature system, varies between –15°C and -5°C, and for a low-temperature system, the evaporation temperatures are in the range of –30°C to -40°C. Variations in temperature are dependent upon products, display cases and the chosen refrigeration system [LIT96].

2.3 Heat load, refrigeration capacity

The heat load of the case is the amount of heat that must be removed from the display case in order to maintain the product in the case at the desired storage temperature. The refrigeration capacity is equal or superior to the heat loads to maintain the product temperature. The refrigeration capacity of a display case is most often given at a specific blown air temperature at the outlet of the evaporator, since this value is easier to measure (and control) than the temperature of the stored product. The standard rating condition to specify the refrigeration capacity of a display case is for operation in an indoor environment with a 75°F dry-bulb temperature and a relative humidity of 55 percent. The heat loads of a refrigerated cabinet are coming from convection, conduction, radiation, and advection.

2.3.1 Conduction

Ambient heat that passes through the walls of the display case is intercepted by the air flowing around the perimeter of the display case.

2.3.2 Radiation

Thermal radiation heat transfer occurs between the interior of the display case and the surrounding ambient environment.

2.3.4 Convection (air entrainment)

The air curtain passing across the opening of the display case mix with and entrain part of the surrounding ambient air, which is then returned to the case evaporator. The heat load due to the entrained air consists of both sensible and latent heats. Ambient air entrainment occurs in all display case types, but represents the largest portion of the refrigeration load for open, multi-deck cases.

2.3.4 Internal loads

Heat energy is generated by the use of electric energy in the display case for the following auxiliaries:
- Lights: fluorescent light features are installed in the display cases for illumination of the product. Heat from the ballasts may also enter the case if the ballast is installed in the refrigerated portion of the case.

- Fan motors: the electric energy associated with the fans used to circulate air around the display case.
- Anti-sweat heaters: are installed in glass doors and on other surfaces that operate at a temperature below the ambient dew-point temperature. If heaters are not installed, condensation and possibly frost will form on these surfaces.

    The contribution of each load source will vary according to display case type. The refrigeration load of open multi-deck display cases is dominated by air entrainment. Internal electric loads represent a significant portion of the refrigeration load of reach-in frozen food cases. For single-deck and tub cases, radiation heat transfer accounts for a large fraction of the heat loads.

    The impact of each of these thermal loads on the refrigeration capacity depends upon the case type. For example, air infiltration is the most significant portion of heat loads for open, multi-deck cases, while radiation is the largest part of the heat load for tub-type cases. The door anti-sweat heaters represent a major share of the refrigeration load for frozen food door reach-in cases.

- Defrosting: The conditions of air in cold storage rooms or in display cases affect the refrigerating capacity of the coil. At surface temperature lower than the dew point temperature of the air, the water vapor contained in the humid air will condense on surfaces, and at surface temperature lower than 0°C frost will deposit on the surfaces. The frost formation that is seen on evaporator surfaces is an important factor in the operation of refrigeration systems. Without periodic removal, the frost will accumulate and eventually block the airflow passages of the evaporator, resulting in loss of cooling capacity. The usual operation for supermarket refrigeration systems is to defrost the display cases on a scheduled basis. Several different methods are employed for defrosting: off-cycle defrosting, electric defrosting.

2.3.5 Off-Cycle defrosting

Refrigeration to the case is shut off and the evaporator warms above the melting temperature of the frost. This method is commonly used for display cases operating at the highest blown air temperatures (34 to 37°F), because frost loading is relatively small. Off-cycle defrosting is also used where the product is not sensitive to air temperature change, such as milk and other dairy products. For frozen food or meat, off-cycle defrosting is not appropriate.

2.3.6 Electric defrosting

Electric heaters are installed at the inlet of the evaporator so that the circulated air can be heated. The warm air passes through the evaporator where it provides the heat needed to melt the frost. Although it is the most energy consuming application, electric defrosting remains used in all refrigeration systems and is considered the most reliable defrosting method.

Defrosting has a significant impact both for energy consumption and product temperatures because of the air and product temperature rises during defrosting and has to be lowered quickly after defrosting leading to a significant overcapacity for rapid "pull down" of temperatures. If not performed correctly, the product can be damaged. The number of defrosting cycles required for a refrigeration case depends on its type. Open, multi-deck display cases will require several, while tub and reach-in cases normally have only one defrosting per day. Defrosting schedule is normally controlled by a time clock that initiates defrosting for each case at specific times each day.

2.4   Thermal modeling of display cases

The average air temperature, inlet and return air temperatures, evaporating temperature, electrical data for fans, heating wires, defrosting heaters and light, coil volume, diameter of tubs and refrigeration loads at 22°C – 65% RH and at 25°C – 60% RH for each cabinet have been put into a database. The refrigeration loads in display cases are dependent on indoor conditions in the supermarket; a higher indoor temperature and relative humidity increase the cooling demand and the energy requirement. An energy balance of an open vertical cabinet is shown in Figure 2.2 where heat losses from infiltration, radiation, conduction, lighting, the fan, heating wires, and defrosting are presented.
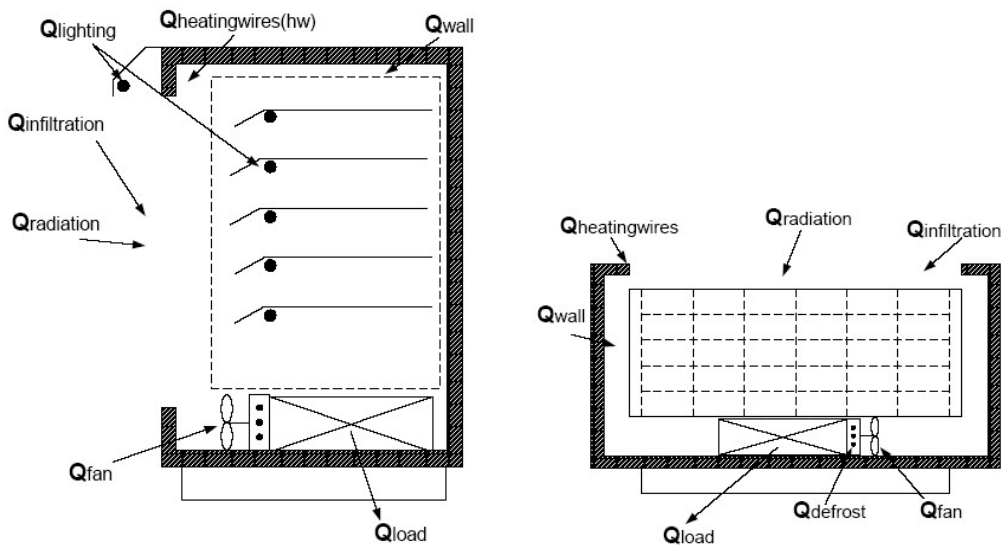
Figure 2. 2 Energy balance of vertical (left) and horizontal (right) display cases.

The following equations state the expressions of different loads accounted for in the cooling load calculations as a function of the display case and the store temperature.

Starting with the conduction load expressed in Equation 2.1:

$$\dot{Q}_w = U_{case} A_{case} \left(T_{store} - T_{case} \right) \tag{2.1}$$

Where $U_{case} = \dfrac{1}{\dfrac{1}{h_i} + \dfrac{t}{k} + \dfrac{1}{h_o}}$ and A_{case} corresponds to the total surface of the cabinet exchanging by conduction with the surrounding store ambience. h_i and h_o represent the inner and outer convective heat exchange coefficients respectively, t and k the insulation thickness and thermal conductivity respectively.

The radiation load is also considered and can be evaluated applying Equation 2.2:

$$\dot{Q}_{radiation} = \frac{\sigma\left(T_w^4 - T_{case}^4\right)}{\left[\left(\dfrac{1-\varepsilon_w}{\varepsilon_w A_w}\right) + \left(\dfrac{1}{A_w F_{case,w}}\right) + \left(\dfrac{1-\varepsilon_{case}}{\varepsilon_{case} A_{case}}\right)\right]} \tag{2.2}$$

Where the subscript w refers to the store wall, σ Stefan Boltzmann's constant and ε the surface emissivity.

The infiltration load depends on the amount of store air entrained in through-frozen-food cabinets. This amount is usually expressed as a ratio of the cabinet blown airflow rate. The percentages of the store air entrapment into the cabinets and freezer rooms are found in literature or evaluated through extensive measurements and parametric analyses for specific blown air velocities. For instance, for an open cabinet, air entrapment is taken equal to 7 or 8% of the blown airflow rate, while it is approximately equal to 1% for closed cabinets. Equation 2.3 states the infiltration load expression:

$$\dot{Q}_{infiltration} = \rho_{air} V_{air,ent} \left(h_{store} - h_{case}\right) \tag{2.3}$$

Where the subscript air,ent refers to entrapped air, V the volumetric air flow rate and h air enthalpy at store or case temperature.

Dissipations of heat from installed equipment should also be taken into account such as lamps and ballasts ($\dot{Q}_{lighting}$), fan motors ($\dot{Q}_{fan}$), anti-sweat heaters ($\dot{Q}_{heating-wires}$) and extra heat from defrost ($\dot{Q}_{Defrost}$). The total load is obtained by summing all of the above evaluated quantities as expressed in Equation (2.4):

$$\dot{Q}_{load} = \dot{Q}_{wall} + \dot{Q}_{infiltration} + \dot{Q}_{radiation} + \dot{Q}_{lighting} + \dot{Q}_{fan} + \dot{Q}_{heating-wires} + \dot{Q}_{Defrost} \tag{2.4}$$

The load calculation of a stand-alone equipment is identical to a closed cabinet display load calculation.

2.5 Thermal modeling of cold storage room

The capacity demand for cold storage is due to four factors: heat transmission, exchange of air, cooling or freezing of products and internal heat generation [GRA03]. Heat transmission through walls, floor, and ceiling is dependent on the overall heat transfer coefficient and the temperature difference between the room and the surroundings. The heat transmission has been defined as shown in Equation 2.5:

$$\dot{Q}_{cond} = \Sigma\left(U_{CR} A_{CR} \left(T_{store} - T_{CR}\right)\right) \tag{2.5}$$

The exchange of air in cold rooms depends on the frequency of door openings and the size of the room. The exchange of air increases the refrigeration load of the room. The influence of incoming air in the room can be calculated from Equation 2.6:

$$\dot{Q}_{airex} = \rho_{air}\dot{V}_{airex}\left(h_{store} - h_{CR}\right) \tag{2.6}$$

Where $\dot{V}_{airex}$ is an average volume flow of incoming air that is defined in (Granryd 2003) as presented in Equation 2.7:

$$\dot{V}_{airex} = V_{CR}\frac{nd}{24.3600} \tag{2.7}$$

nd is the number of air exchanges in the room per 24 hours. Temperatures and the frequency of door openings influence the number of air exchanges. Results from experiments are presented in Table 2.1 [GRA03].

Table 2. 1 Number of air exchanges from ARIAS.

Room Volume (m³)	Medium T(°C)	Low T(°C)
7	38	30
10	31.5	24.5
20	21.5	17
40	14.5	11.5
100	9	7
500	3.5	2.7
1000	2.5	2.7
3000	1.35	1.05

The enthalpy difference for freezer rooms has been assumed to be 45 [kJ/kg], which is the average between the enthalpies of different products at temperatures -15°C and –18°C. Similarly, the enthalpy difference for the cold room has been assumed to be 55 [kJ/kg], which is the average between the enthalpies of different products at temperatures 17°C and 1°C. The mass flow has been assumed to be 20 kg/m³ per 24 hours [ARIA05] for cold rooms and 15 kg/m³ per 24 hours for freezer rooms. Internal heat generation from lighting and people also affects the refrigeration load of the cold room. The heat generated by lighting has been assumed to be 15 W/m² and the heat from people to be 200 W.

2.6   Coefficient of performance

Many factors have an impact on the coefficient of performance (COP):
 - Level of temperature for product or beverage conservation
 - Temperature differences at the condenser and evaporator coils
 - Compressor efficiency
 - Type of refrigerant
 - Configuration of the refrigeration system (one or two compression stage, sub-cooling or not)

For energy calculation along the year, the coefficient of performance has been considered as a function of these variable values. The expression of the COP for the theoretical cycle of Carnot is:

$$COP_c = \frac{Te}{Tc - Te} \tag{2.8}$$

Where *Te* and *Tc* are respectively evaporating temperature and condensing temperature expressed in Kelvin. These temperatures are linked to product temperature and ambient temperature. Product conservation temperature is supposed to be fixed along the year. Typical temperature difference at the evaporator and the condenser are presented in Table 2.2.

Table 2. 2 Temperature difference in heat exchangers.

Difference of temperature in heat exchangers	DTev evaporator	DTcd condenser
Centralized System / medium temperature	15 K	12 K
Centralized System / low temperature	17 K	10 K
Condensing Units / medium temperature	15 K	12 K
Condensing Units / low temperature	17 K	12 K
Stand-alone equipment / medium temperature	15 K	15 K
Stand-alone equipment / low temperature	15 K	15 K

In centralized systems, the evaporating temperature is the same for all display case connected to the same compressor rack. The evaporating temperature varies usually between −10°C and −12°C for medium temperature display cases. Product temperature conservation ranges from 0°C to +10°C. Moreover, the control for evaporator feeding in refrigerant is usually performed by an electromagnetic valve. The superheat control is not optimized and leads to poor efficiency of the evaporator coil. In consequence, temperature difference between air and refrigerant is high.

Because of pressure drops in suction line of the compressor rack, the pressure at the suction port is lower, and the equivalent saturating temperature is between −12°C and −15°C. The air flow rate on the evaporator is low, for noise reduction.

For the calculations, the temperature chosen is **−13°C** for medium-temperature compressor rack, and −35°C for low-temperature compressor rack.

The coefficient of performance can be expressed with the theoretical Carnot COP and the cycle efficiency. The cycle efficiency depends mainly on the compressor efficiency, the cycle architecture, and the refrigerant properties. For each type of refrigeration system (centralized, condensing units, stand alone), cycle efficiencies have been calculated for the refrigerant in use. Compressor efficiencies have been taken from manufacturer data (Copeland, Carlyle)

Table 2. 3 Cycle efficiency (COP / COPc).

Refrigeration system	Cycle efficiency
Centralized System / medium temperature	45 %
Centralized System / low temperature	42 %
Condensing Units	40 %
Stand-alone equipment	25 %

The additional power consumption of the condenser fan is integrated in the cycle efficiency. In centralized systems, the input power for condenser ventilation is 7% of compressor rack input power for medium-temperature system, and 8% of compressor rack input power for low-temperature system [BIG02, FAY00].

In supermarkets, the condensation pressure is controlled to a minimum level, to keep the pressure sufficiently high in order that the thermo expansion valves (TxV) feed correctly the evaporators. In winter time, with low outdoor temperature, it is possible to reduce the condensing

pressure, taking advantage of a lower pressure ratio for the compressors. This is possible when expansion valves are designed for a wider range of pressure differences. The impact on the cycle efficiency, when the head pressure control is activated, has been taken into account for the different scenarios of energy consumption.

Finally the equation of the coefficient of performance is a function of the outdoor temperature:

$$COP = \frac{Tev}{(Text + DT_{cd}) - Tev} \cdot \eta c \qquad (2.9)$$

with *DTcd* (temperature difference at the condenser) as functions of the out door temperature (*Text*), the floating head pressure control (*FHPC*), the technology of the refrigerating system (*Tech*), and the level of the evaporating temperature (*Tev*) (see Table 6).

$$DTcd = f(Text; FHPC; Tech; Tev) \qquad (2.10)$$

The cycle efficiency (η_c) is a function of the technology (*Tech*) and the level of temperature for centralized systems (*Tev*) (see Table 7). Two other variables are considered for the cycle efficiency: the floating head pressure control (*FHPC*) and the outdoor temperature (*Text*).

Because ventilation is reduced when outdoor temperature is low, in winter time for example, the additional input power of the condenser fan decreases. The cycle efficiency includes the additional power input of the condenser fans, which varies according to the outdoor temperature. In consequence, the cycle efficiency must be correlated to the outdoor temperature and the head pressure control.

$$\eta c = f(Text; FHPC; Tech; Tev) \qquad (2.11)$$

2.7   Store distribution in climatic zones

Global calculations for energy consumption are done for different climatic conditions. The outdoor temperature has an impact on the coefficient of performance of the refrigerating system (8), (10). Depending on the climatic zone of the supermarket and the other stores, the hourly variation of temperature will have an impact on the energy consumption.

8 climatic zones have been defined. Temperature variations during one typical year are known and registered hour by hour in weather stations. The distribution of stores and supermarkets in the different climatic zones has been done proportionally to the population living in each zone.

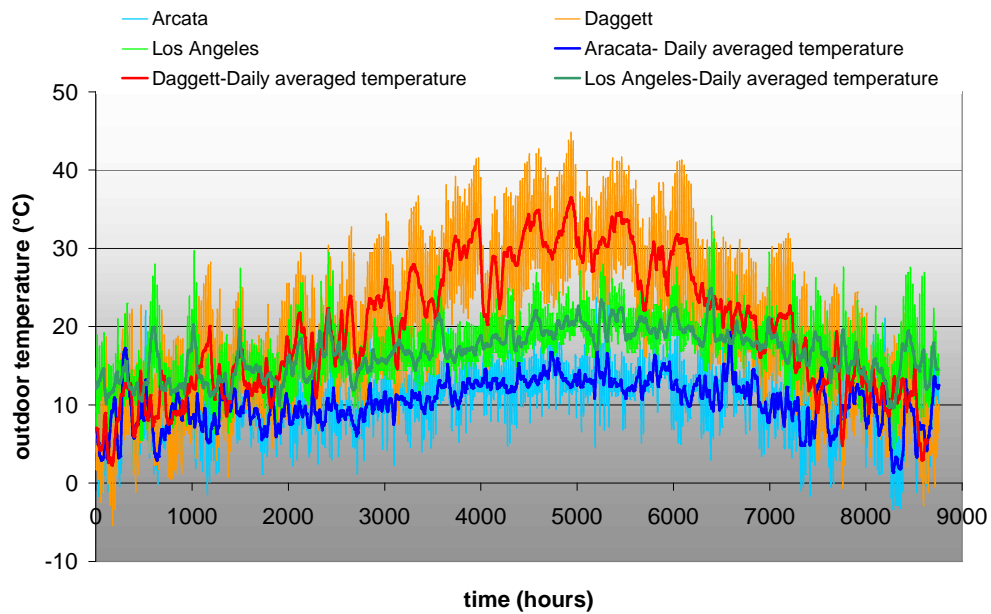Figure 2.3 presents the average temperature (24 hours averaged) for 3 weather stations in California.

Figure 2. 3 Temperature evolution in one year – 3 weather station measurements.

Figure 2.4 presents the 8 climatic zones, and the population distribution in these zones. Los Angeles, with 36% of the population, is the first for the number of inhabitants. The distribution of the different stores studied in the commercial refrigeration sector is supposed to be the same as the population distribution.
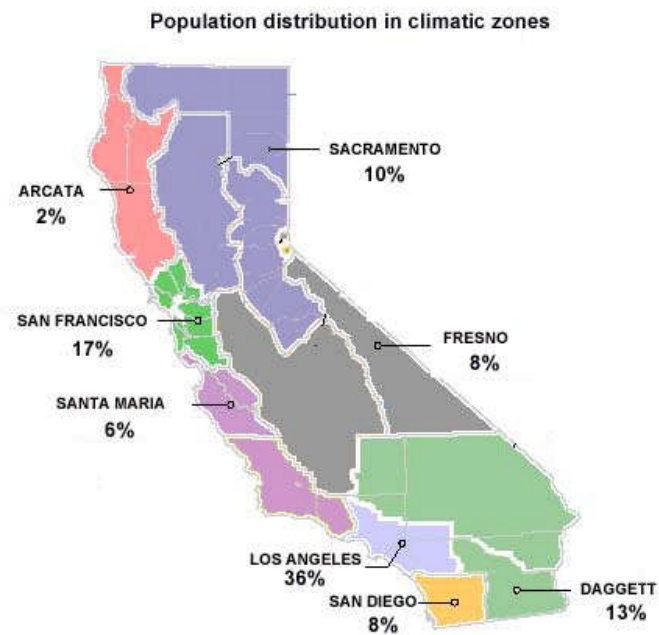

Figure 2. 4 Climatic zones and population distribution.

3   Energy Savings

The quest for increased energy efficiency and the phase-out of ozone depleting substances (ODS) have changed the refrigeration system design for some new supermarkets. A great potential for energy efficiency improvement as well as limitation of green House gas emissions have been assessed by many research laboratories as well as commercial chains or equipment manufacturers. Energy saving technologies such as heat recovery, floating head pressure, defrosting control, energy efficient lighting, high efficiency motors, and efficient control have been implemented in many supermarkets to reduce energy consumption. The objective of these technical options is to develop energy-efficiency solutions in refrigerated cases, while enhancing food safety without hampering merchandizing facets.

3.1   Heat recovery systems

The refrigeration system in a supermarket always rejects heat to the environment through the condensers. It is possible to use the rejected heat for heating the store in the winter season. Heat rejected from condensers can also be used to heat service water and premises in cold climates, which is a good measure to improve energy usage in new efficient refrigeration cycle design. Heat recovery leads to a reduction in costs and in the usage of fossil fuels for heating.

3.2   Floating head pressure

A drawback of the heat recovery system is the high condensing temperatures that increase the energy consumption of the refrigeration system.   In the so-called floating head pressure condensing systems, the condensing temperature follows with the ambient temperature. The system is implemented with electronic expansion valve (or multiple-orifice expansion valve) operating over a wide range of pressure differences and allowing for low condensing temperature at low ambient temperatures. A reduction of condensing temperatures increases the coefficient of performance of refrigeration systems.

3.3   Installing glass doors in open cases

Display cases commonly carry large refrigeration loads, especially vertical open display cabinets. The reason is that this kind of cabinet displays a large amount of food on a small surface in the store with a large open front area. The heat and moisture exchanged between the products in the cabinet and the store environment affect the refrigeration load, defrost and condensation, on walls and products. Infiltration causes about 60 to 70% of the cooling load for a typical open vertical display cabinet [ARIA05].

The refrigeration loads associated with the glass door reach-in case are normally less than those of the multi-deck, but greater than for the tub case. Glass door cases are, however, equipped with anti-sweat electric heaters in the doors to prevent fogging and decreased visibility of the product.

Installing glass doors in display cases reduces the infiltration and energy consumption of the cabinets. The reason for the absence of the doors in a display case is to avoid placing an obstacle between the customer and the product, which may hinder the customer impulse to

purchase a new product. Results from a laboratory test that evaluated glass doors on a open five-deck display case show a reduction of the total cooling load of the case by a 68% [FAR02].

3.4   Hot Gas Defrost

Discharge refrigerant gas is piped from the compressor rack to the display case where the refrigerant is condensed by melting the frost. The piping is arranged so that the liquid refrigerant is returned to the compressor rack for distribution to other display cases in the system. Hot gas defrosting is the fastest method to remove frost and tends to have the least impact on case air and product temperatures. Hot gas is the most costly defrosting method to implement because of the extensive piping and controls needed.

4 Results for energy consumption

4.1 Energy consumption in the commercial refrigeration sector

Results for energy consumption are presented first for supermarkets only, and second for all the commercial refrigeration sector, including small stores and vending machines.

4.1.1 Results for grocery supermarkets

One typical grocery supermarket

Before deriving the calculation for California State, one typical supermarket located in the Los Angeles (LA) climatic zone is presented. Table 4.1 gives the cooling capacity distribution, for medium and low temperature systems, and for each technology of display cases and walk-in coolers.

Table 4. 1 Cooling capacity in a typical grocery supermarket.

Cooling Capacity	Medium temperature	Low temperature	Total
Centralized System (kW)	193	152	345
Condensing Units (kW)	15	11	28
Stand-alone (kW)	11	1	12
Total (kW)	**219**	**164**	**385**

90% of the total cooling capacity of refrigeration equipment is the centralized system. Stand-alone display cases totalize 12 kW of cooling capacity, which is 3 % of the refrigeration capacity in the grocery supermarket.

Refrigerant charge is around 1600 kg including stand-alone equipment and condensing units. Centralized system represents 90% of the total amount. The evaluation of the refrigerant charge is presented in details in section 7 for refrigerant emission inventory.

Energy consumption for one supermarket in LA climatic zone

Energy consumption is calculated hour by hour, for each climatic zone. Results for one supermarket in LA climatic zone are presented in Table 4.2.

Table 4. 2 Annual energy consumption for 1 grocery supermarket in LA climatic zone.

Grocery supermarket (GWh/year)	Centralized System	Condensing Units	Stand-alone
Compressor for refrigeration	0.827	0.098	0.041
Auxiliary components	0.504	0.071	0.027
AC additional energy consumption	0	0	0.013
Total (GWh)	1.331	0.169	0.081
		1.581 GWh	

The energy consumption of refrigeration compressors is 0.827 GWh/year. Auxiliary components (fans, lighting, anti-sweat heaters, and defrosting heaters) totalize 0.504 GWh. As mentioned by different studies, (Wal03, Bax03, ORL04, Lit96) field tests on energy consumption measurement in a supermarket have concluded on numbers in the same order of magnitude.

Derivation to California State

Taking into account the different climatic zones, and the distribution of the stores in these zones, the energy consumption is derived to California (Table 4.3).

Table 4. 3 Annual energy consumption for grocery supermarkets in California.

Grocery supermarkets in CA (GWh/year)	Centralized System	Condensing Units	Stand-alone
Compressor for refrigeration	2,810	334	137
Auxiliary components	1,692	237	92
AC additional energy consumption	0	0	39
Total (GWh)	4,502	571	268
	5,341 GWh		

Annual energy consumption of commercial refrigeration equipment in grocery supermarkets, including auxiliary electric loads, is 5,341 GWh in California. 84% is due to centralized systems, and 5% of the total energy consumption is due to stand-alone display cases.

4.1.2 Results including the other small stores using refrigeration equipment

Table 4. 4 Annual energy consumption for commercial refrigeration sector in California, small stores included.

Total commercial ref. (GWh/year)	Centralized System	Condensing Units	Stand-alone
Compressor for refrigeration	2,810	3,831	4,548
Auxiliary components	1,692	2,364	3,802
AC additional energy consumption	0	0	1,181
Total (GWh)	4,502	6,196	9,531
	20,228 GWh		

When all types of stores are added to grocery supermarkets, the annual energy consumption grows to 20,228 GWh. The share by technology of refrigeration equipment is presented on Figure 4.1.
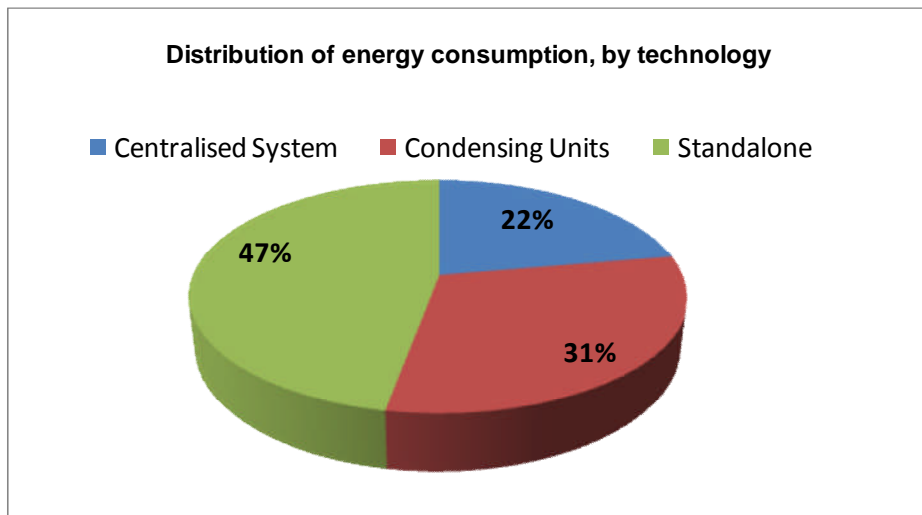
Figure 4. 1 Distribution of energy consumption by technology, in commercial refrigeration

Because of the high number of stand-alone equipment in small stores, including vending machines, and the poor efficiency of their refrigerating systems, this technology is first for energy consumption in the commercial refrigeration sector. It costs approximately 10 TWh per year in California. Centralized systems, only used in supermarkets, are more energy efficient and represent 22% of the global energy consumption. But in terms of refrigerant bank and refrigerant emissions, the situation is reversed and centralized systems are responsible for nearly 80% of refrigerant emissions in the commercial refrigeration sector.

4.2 Technical options for energy savings

5 technical options for energy savings have been evaluated.

- **Technical option 1:** night curtain are installed on each open display case. The ambient air induction is reduced, and the thermal load on the refrigeration system decreases. Night hours have been considered from 10 pm to 4 am. Moreover, during night hours, lighting is off in all display cases.
- **Technical option 2:** all open display cases are replaced by glass door display cases, even for medium temperature products. Ambient air induction is significantly reduced (by factor 7), decreasing the thermal load of the display case and the energy consumption of the refrigeration system.
- **Technical option 3:** auxiliary components are replaced by new technologies, with improved energy efficiency (LED lighting, DC current fan, high efficiency heater…)
- **Technical option 4:** the floating head pressure control is done on every centralized system in supermarkets. Depending on the climatic zone, the impact is more or less significant on the annual energy consumption.
- **Technical option 5:** all options combined: 100% glass door + high efficiency electrical components + floating head pressure control.

4.2.1 Technical option 1: Night curtain installed on every open display case

- Grocery supermarkets

Table 4.5 presents the results for one typical supermarket in LA climatic zone. Energy savings, thanks to night curtains installed on every open display case in a supermarket is 92 MWh/year, 5.82% of the energy consumption without night curtain.

Table 4. 5 Energy consumption for one grocery supermarket – Technical option 1

Operating Mode	Night curtains		
Grocery supermarket (GWh/year)	**Centralized System**	**Condensing Units**	**Stand-alone**
Compressor for refrigeration	0.745	0.096	0.038
Auxiliary components	0.497	0.072	0.028
AC additional energy consumption	0	0	0.013
Total	1.242	0.168	0.079
	1.489		
Energy Savings (GWh ; %)	0.092	5.82%	

Operating Mode	Night curtains		
Grocery supermarkets in CA (GWh/year)	*Centralized System*	*Condensing Units*	*Stand-alone*
Compressor for refrigeration	2,533	327	128
Auxiliary components	1,666	240	94
AC additional energy consumption	0	0	37
Total	4,200	567	259
		5,027	
	Energy Savings (GWh ; %)	313	6%

Table 4. 6 Energy consumption for all supermarkets in California – Technical option 1.

Deriving the calculation for all supermarkets in California, the energy saving associated to the installation of night curtain is 313 GWh/year.

- All commercial refrigeration equipment (small stores and supermarkets) in California

Considering now all refrigeration equipment, the additional savings is limited (313 to 351 GWh), because most of stand-alone equipment is already closed with glass doors and the night curtain technical option has no effect on this equipment.

Table 4. 7. Energy consumption in commercial refrigeration sector – Technical option 1.

Operating Mode	Night curtains		
Total commercial ref. (GWh/year)	*Centralized System*	*Condensing Units*	*Stand-alone*
Compressor for refrigeration	2,534	3,797	4,532
Auxiliary components	1,666	2,365	3,805
AC additional energy consumption	0	0	1,178
Total	4,200	6,162	9,515
		19,877	
	Energy Savings (GWh ; %)	351	2%

4.2.2 Technical option 2: Open display cases closed with glass doors

The impact of night curtain is limited to 6 hours in the night. Moreover this period of time is one where coefficient of performance increases thanks to quite low outdoor temperatures. A radical change in display case technology could be to close all open display cases with doors, medium-temperature ones included.

- Grocery supermarkets

Table 4. 8 Energy consumption for one grocery supermarket – Technical option 2.

Operating Mode	Add doors		
Grocery supermarket (GWh/year)	*Centralized System*	*Condensing Units*	*Stand-alone*
Compressor for refrigeration	0.539	0.084	0.034
Auxiliary components	0.611	0.074	0.031
AC additional energy consumption	0	0	0.011
Total	1.15	0.158	0.076

		1.384	
Energy Savings (GWh ; %)		0.197	12.46%

Closing all the display cases, the energy savings for one year is 12.5%: 200 MWh per supermarket

Table 4. 9 Energy consumption for all supermarkets in California – Technical option 2.

Operating Mode	Add doors		
Grocery supermarkets in CA (GWh/year)	*Centralized System*	*Condensing Units*	*Stand-alone*
Compressor for refrigeration	1,812	281	114
Auxiliary components	2,050	250	105
AC additional energy consumption	0	0	32
Total	3,862	531	251
	4,644		
Energy Savings (GWh ; %)		697	13.05%

Deriving the scenario to California, the energy saving is nearly 0.7 TWh per year.

- All commercial refrigeration equipment (small stores and supermarkets) in California

Table 4. 10 Energy consumption in commercial refrigeration sector – *Technical option 2.*

Operating Mode	Add doors		
Total commercial ref. (GWh/year)	*Centralized System*	*Condensing Units*	*Stand-alone*
Compressor for refrigeration	1,812	3,412	4,518
Auxiliary components	2,050	2,374	3,816
AC additional energy consumption	0	0	1,173
Total	3,862	5,786	9,507
	19,155		
Energy Savings (GWh ; %)	1,073		5.305%

Most of stand-alone equipment is already equipped with glass doors. The impact on energy savings is significant mainly in supermarkets. Nevertheless, for the complete commercial refrigeration sector, the energy savings are 5.3 % compared to the base line.

4.2.3 Technical option 3: Cabinet lighting, anti-sweat heater and ventilation: low energy consuming technologies

- Grocery supermarkets

Table 4. 11 Energy consumption for one grocery supermarket – Technical option 3

Operating Mode	Eco for auxiliary components		
Grocery supermarket (GWh/year)	*Centralized System*	*Condensing Units*	*Stand-alone*
Compressor for refrigeration	0.698	0.078	0.038
Auxiliary components	0.358	0.047	0.023
AC additional energy consumption	0	0	0.013
Total	1.056	0.125	0.074
	1.255		

Energy Savings (GWh ; %)		0.326	20.62%

Auxiliary components are energy consumers, first by their own electrical load, and second by the additional heat load to the display case. This additional heat load increases the cooling capacity of the refrigeration system, and its energy consumption.

Improved technologies are available to retrofit lighting, ventilation, and anti-sweat heaters. Technical option 3 gives the range of energy savings if all auxiliary components were replaced.

Table 4. 12 Energy consumption for all supermarkets in California – Technical option 3.

Operating Mode	Eco for auxiliary components		
Grocery supermarkets CA (GWh/year)	*Centralized System*	*Condensing Units*	*Stand-alone*
Compressor for refrigeration	2,345	261	128
Auxiliary components	1,200	159	77
AC additional energy consumption	0	0	37
Total	3,545	420	241
		4,206	
Energy Savings (GWh ; %)		1,134	21.24%

Energy savings thanks to high efficiency auxiliary components is around 15% compared to the base line. In California, the annual savings are 1.2 TWh for this technical option.

- All commercial refrigeration equipment (small stores and supermarkets) in California

Table 4. 13 Energy consumption in commercial refrigeration sector – Technical option 3.

Operating Mode	Eco for auxiliary components		
Total commercial ref. (GWh/year)	*Centralized System*	*Condensing Units*	*Stand-alone*
Compressor for refrigeration	2,345	3,058	4,112
Auxiliary components	1,200	1,488	3,125
AC additional energy consumption	0	0	1,062
Total	3,545	4,546	8,298
		16389	
Energy Savings (GWh ; %)		3,840	19.0%

Technical option 3 is applied to each type of display cases. Stand-alone equipment can benefit of the technical changes. Deriving the scenario to California, for commercial refrigeration sector, energy savings for one year are 7.85 TWh, nearly 19% of the base line consumption.

4.2.4 Technical option 4: Floating head pressure on centralized systems

- Grocery supermarkets

Table 4. 14 Energy consumption for one grocery supermarket – Technical option 4.

Operating Mode	Floating head pressure (Eco)		
Grocery supermarket (GWh/year)	*Centralized System*	*Condensing Units*	*Stand-alone*
Compressor for refrigeration	0.778	0.089	0.041
Auxiliary components	0.504	0.071	0.027
AC additional energy consumption	0	0	0.013

	Total	1.282	0.16	0.081
			1.523	
	Energy Savings (GWh ; %)		0.058	3.67%

Table 4.14 presents the energy consumption for one supermarket in LA climatic zone, when floating head pressure control is activated. The interest of this control, and the energy savings associated are strongly dependent on the temperature changes during the year. In a climatic zone where maximum and minimum temperatures are not far from each other, the interest of a floating head pressure is limited. In LA climatic zone, the energy saving is 3.7%.

The derivation of energy consumption in California, taking into account 8 climatic zones, give a better result with 5% of energy savings thanks to the floating head pressure control.

Table 4. 15 Energy consumption for all supermarkets in California – Technical option 4.

Operating Mode	Floating head pressure (Eco)		
Grocery supermarkets in CA (GWh/year)	*Centralized System*	*Condensing Units*	*Stand-alone*
Compressor for refrigeration	2,583	296	137
Auxiliary components	1,692	237	92
AC additional energy consumption	0	0	39
Total	4,275	534	268
	5,077		
Energy Savings (GWh ; %)		264	5%

- All commercial refrigeration equipment (small stores and supermarkets) in California

Table 4. 16 Energy consumption in commercial refrigeration sector – Technical option 4.

Operating Mode	Floating head pressure (Eco)		
Total commercial ref. (GWh/year)	*Centralized System*	*Condensing Units*	*Stand-alone*
Compressor for refrigeration	2,583	3,400	4,548
Auxiliary components	1,692	2,364	3,802
AC additional energy consumption	0	0	1,181
Total	4,275	5,764	9,531
	1,9570		
Energy Savings (GWh ; %)		658	3.251%

Floating head pressure cannot be applied to stand-alone equipment, which are located in an air conditioned area. The global impact of this technical option, on complete commercial refrigeration sector is lowered to 3.3%, representing 0.66 TWh per year.

4.2.5 Technical option 5: All options combined

The last scenario is the combination of three factors: closing all open display cases, low energy consuming components, and floating head pressure.

- Grocery supermarkets

In a typical supermarket, located in LA climatic zone, the annual energy savings is 37%, meaning 811 MWh less consumed (see Table 4.17).

Table 4. 17 Energy consumption for one grocery supermarket – Technical option 5.

Operating Mode	Add doors + eco Aux + FHP		
Grocery supermarket (GWh/year)	*Centralized System*	*Condensing Units*	*Stand-alone*
Compressor for refrigeration	0.489	0.071	0.031
Auxiliary components	0.442	0.05	0.026
AC additional energy consumption	0	0	0.01
Total	0.931	0.121	0.067
	1.119		
Energy Savings (GWh ; %)		0.462	29.22%

Table 4. 18 Energy consumption for all supermarkets in California – Technical option 5.

Operating Mode	Add doors + eco Aux+Eco FHP		
Grocery supermarkets in CA (GWh/year)	*Centralized System*	*Condensing Units*	*Stand-alone*
Compressor for refrigeration	1,621	235	103
Auxiliary components	1,482	168	86
AC additional energy consumption	0	0	29
Total	3,103	403	218
	3,724		
Energy Savings (GWh ; %)		1,617	30.273%

For all supermarkets in California, the maximum energy savings are 1.62 TWh per year when all technical options are applied.

- All commercial refrigeration equipment (small stores and supermarkets) in California

Table 4.19. Energy consumption in commercial refrigeration sector – Technical option 5.

Operating Mode	Add doors + eco Aux+Eco FHP		
Total commercial ref. (GWh/year)	*Centralized System*	*Condensing Units*	*Stand-alone*
Compressor for refrigeration	1,621	2,955	4,080
Auxiliary components	1,482	1,495	3,135
AC additional energy consumption	0	0	1,052
Total	3,103	4,450	8,267
	15,820		
Energy Savings (GWh ; %)		4,409	21.8%

For the complete commercial refrigeration sector, the maximum energy savings is 4.41 TWh per year, totalizing 22% of the base line consumption. Global annual energy consumption is evaluated at 15.8 TWh, and stand-alone equipment consumes more than half of this value.

4.2.6 Summary

- Energy savings in supermarkets

Figure 4.2 presents the comparison of the energy savings, related to the base line, for different technical options applied to grocery supermarkets in California.
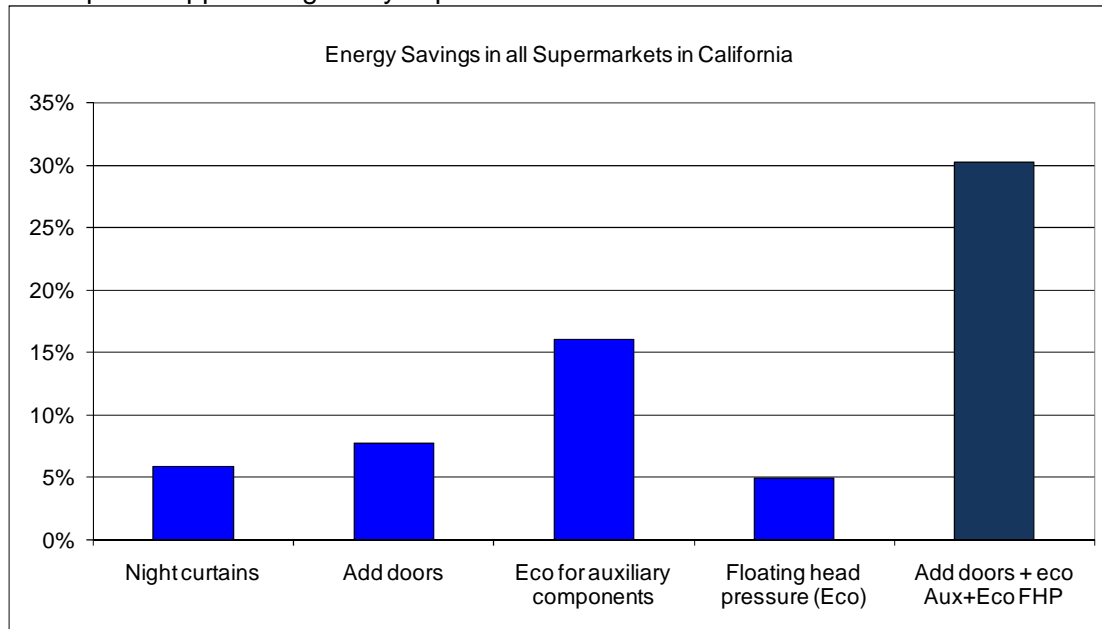
Figure 4.2. Energy savings / technical options applied in Californian supermarkets.

- Night curtains installed on open display cases have a limited impact on the energy consumption. Night period is short in time in supermarkets (6 hours only) and during this period, the coefficient of performance of the refrigerating system is improved thanks to quite low outdoor temperature, lowering the condensation temperature by the same time.
- Floating head pressure control is interesting in climatic zones with wide temperature differences along the year. Near the coast, where the temperature is more stable, the interest of this system is limited.
- In supermarkets, most of display cases are open, and heat loads due to air induction is around 70% of the total load. Closing the display cases permits to decrease the cooling capacity and the energy consumption of the compressor racks. 8 % of energy saving are possible with this change in technology.
- The other elements for energy consumption are the auxiliary components. High energy efficiency technologies exist and could reduce by 16% the energy consumption.
- All options applied together lead to 30% of energy savings in supermarkets

- Energy savings in small stores (condensing units and standalone equipment)

Figure 4.3 presents the comparison of the energy savings, related to the base line, for different technical options applied to small stores in California. Stand-alone equipment and condensing units are the two refrigeration technologies used in small stores.

Energy savings in small stores
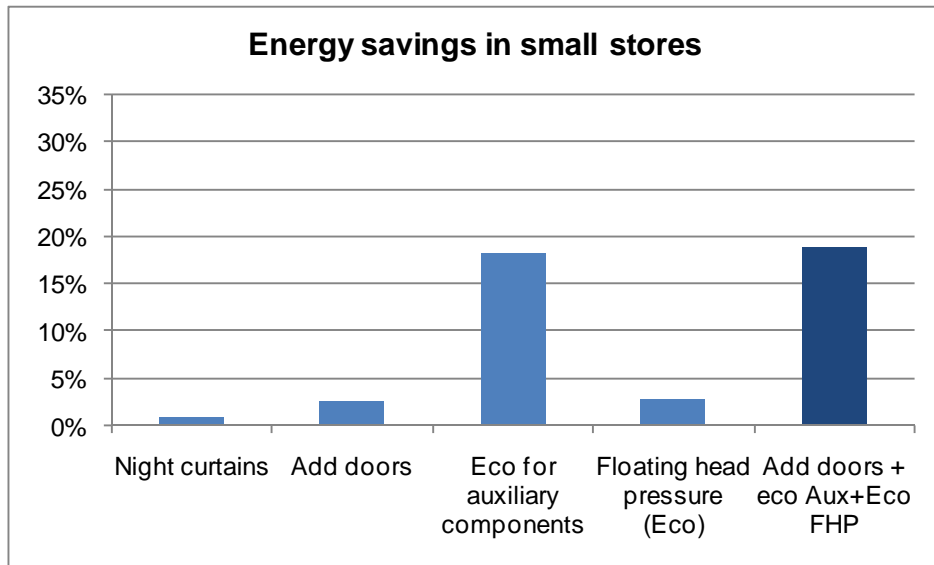
Figure 4.3. Energy savings / technical options applied in Californian small stores.

Most of stand-alone equipment is closed with glass doors (vending machines for example). Options of night curtains and closing the display cases are not applied on this stand-alone equipment. The impact on energy consumption is low.

Progresses to save energy on stand-alone equipment must be focused on auxiliary components and compressor efficiency, which is very poor today.

- Energy savings in commercial refrigeration sector, all types of stores

Figure 4.4 presents the comparison of the energy savings, related to the base line, for different technical options applied to all types of stores using refrigeration equipment in California, whatever the technology.
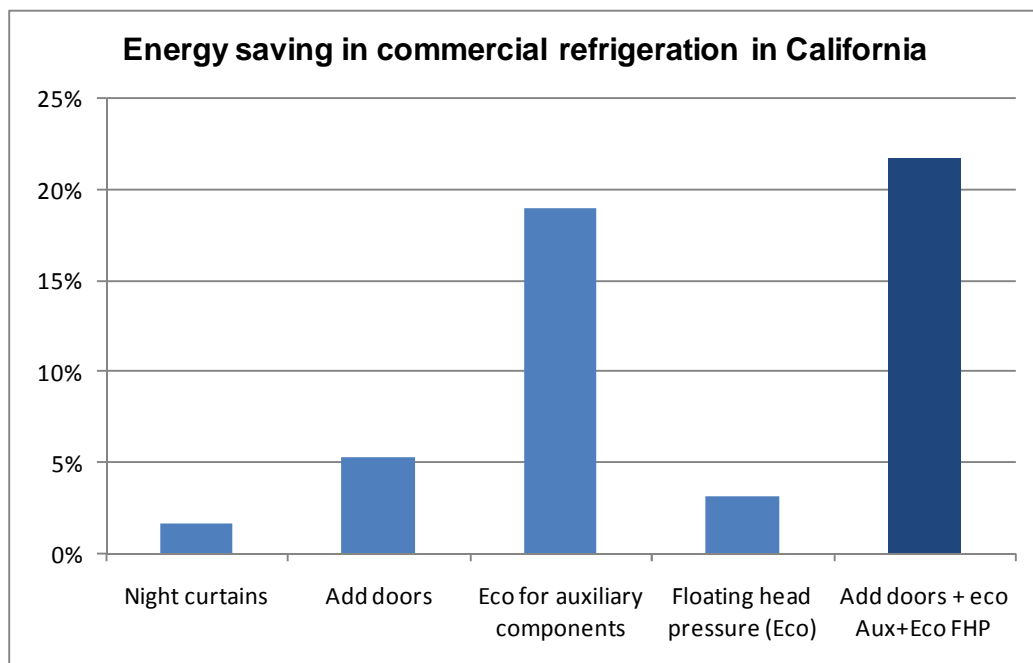
Figure 4.4. Energy savings / technical options applied in commercial sector.

Figure 4.5 presents for technical option 5, where all technical options are applied, the distribution in energy consumption by technology of refrigerating system.
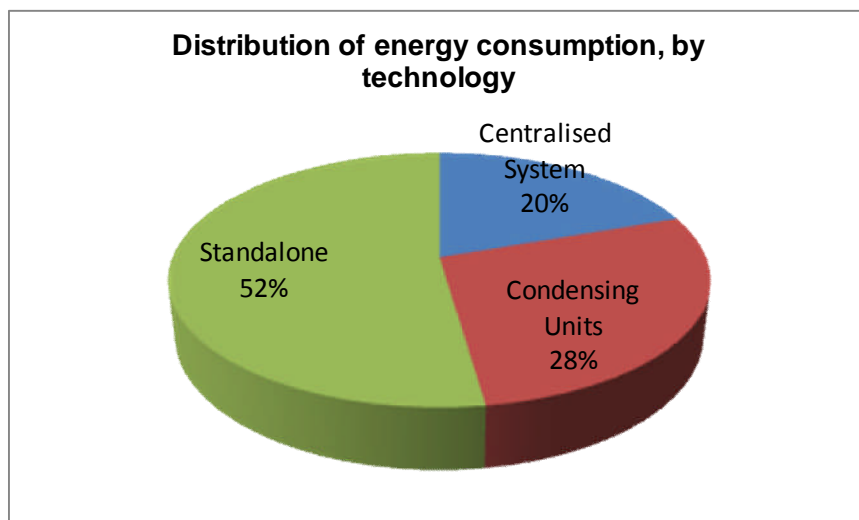
Figure 4.5. All technical options combined: energy consumption distribution.

It appears clearly that stand-alone equipment, by their high numbers in every type of stores, consumes more than 50% of the total energy consumption in commercial refrigeration sector. The poor efficiency of small hermetic compressor, and sometime heat exchanger designs not adapted, lead to a poor cycle efficiency (25%). Technical options to reduce energy consumption are more effective for centralized system. In technical option 5, the energy consumption of centralized systems is cut by nearly 40% compared to the base line.

5   General approach for Life Cycle Cost Analysis

The Life Cycle Cost (LCC) is the total customer cost over the life time of the equipment, including purchase cost and operating cost (including energy cost). Future operating costs are discounted to the time of purchase and summed over the lifetime of the equipment. Inputs to the LCC analysis are categorized as follows:
- inputs for establishing the purchase cost, otherwise known as the total installed cost;
- And inputs for calculating the operating cost (i.e., energy, maintenance, and repair costs).

Life-cycle cost is defined by equation 5.1:

$$LCC = IC + \sum_{t=1}^{N} \frac{OC_t}{(1+r)^t} \tag{5.1}$$

Where LCC = life-cycle cost ($),
IC = total installed cost ($),
N = lifetime of equipment expressed in years,
Σ = sum over the lifetime, from year 1 to year N,
OC = operating cost ($),
r = discount rate,
t = year for which operating cost is being determined.
Because most of data used to conduct the LCC analysis are gathered in 2008, all costs are expressed in 2008 US $.

The LCC analysis is performed for different efficiency levels and LCC difference between the baseline equipment and equipment with higher efficiency level is evaluated. A distribution of LCC differences is then generated to determine the mean LCC difference.

5.1   Total installed cost

The primary inputs for establishing the total installed cost are: the baseline manufacturer selling price, marks up and sales tax and the installation price.

5.2   Baseline manufacturer selling price

Baseline manufacturer selling price is the price charged by the manufacturer to either a wholesaler or customer for equipment meeting existing minimum efficiency (or baseline) standards. The manufacturer selling price includes a markup that converts the cost (i.e., the manufacturer cost) to a manufacturer selling price. Standard-level manufacturer selling price increase: Standard-level manufacturer selling price increase is the incremental change in manufacturer selling price associated with producing equipment at each of the higher standard levels.

5.2.1 Markups and sales tax

Markups and sales tax convert the manufacturer selling price into a customer price.

5.2.2 Installation price

The installation price is the cost to the customer of installing the equipment. The installation price represents all costs required to install an equipment but does not include the marked-up customer equipment price. The installation price includes labor, overheads, and any miscellaneous materials and parts. Thus, the total installed cost equals the customer equipment price plus the installation price and is defined by equation 5.2:

$$IC = EP + InstC \tag{5.2}$$

where EP = equipment price (i.e., customer price for the equipment only), expressed in $, and $InstC$ = the installation cost or the customer price to install equipment (i.e., the cost for labor and materials), also expressed in $.

The equipment price includes the manufacturing cost of an equipment multiplied by different markups. A first markup "the baseline manufacturer markup" converts the cost to manufacture (i.e., the manufacturing cost) to a manufacturer selling price, the price charged by manufacturers to either a wholesaler/distributor or a very large customer for existing equipment. All associated retail markups and applicable sales tax markup together are then multiplied and expressed as the "overall markup". The overall markup in turn is multiplied by a "baseline manufacturer selling price" to attain the price paid by the customer as stated in equation 5.3:

$$EP = OMU \times BMU \times MFC \tag{5.3}$$

Where MFC = manufacturing cost,
BMU = baseline manufacturer markup,
and OMU = overall markup.

The installation cost is the price to the customer of labor and materials (other than the actual equipment) needed to install the refrigeration equipment. Installation costs were derived for commercial refrigeration equipment from data provided by the DOE based on RS Means Mechanical Cost Data.3. RS Means provides estimates on person-hours required to install commercial refrigeration equipment and labor rates associated with the type of crew required to install the equipment [DOE07].

The installation cost is then calculated by multiplying the number of person-hours by the corresponding labor rate. Since labor rates vary significantly from one region to another, the regional variability is taken into account and is expressed in terms of cost indices for 50 states as shown in table 5.1. The total installed cost is therefore expressed as shown in equation 5.4:

$$IC_{CA} = OMU \times BMU \times MFC + InstC_{USA} \times \frac{II_{CA}}{II_{USA}} \tag{5.4}$$

where II represents the cost installation index and CA refers to California. This method is applied to display cases and self contained categories defined in section 1.4.

Table 5.1 Installation cost indices (national value = 100)

State	Index	State	Index	State	Index
Alabama	65.4	Kentucky	73.2	North Dakota	67.0
Alaska	117.1	Louisiana	60.9	Ohio	103.0
Arizona	79.1	Maine	76.9	Oklahoma	67.3
Arkansas	53.7	Maryland	92.1	Oregon	115.3
California	123.8	Massachusetts	123.1	Pennsylvania	128.5
Colorado	88.3	Michigan	112.3	Rhode Island	120.9
Connecticut	111.7	Minnesota	122.8	South Carolina	42.6
Delaware	125.1	Mississippi	41.6	South Dakota	40.1
Dist. of Columbia	97.7	Missouri	104.0	Tennessee	75.2
Florida	64.8	Montana	80.9	Texas	66.7
Georgia	67.3	Nebraska	83.7	Utah	76.6
Hawaii	126.6	Nevada	113.1	Vermont	73.6
Idaho	78.5	New Hampshire	91.9	Virginia	70.8
Illinois	129.1	New Jersey	132.3	Washington	109.8
Indiana	91.7	New Mexico	78.3	West Virginia	93.5
Iowa	85.6	New York	166.3	Wisconsin	99.3
Kansas	75.0	North Carolina	46.4	Wyoming	56.4

5.3   Operating cost

The operating cost includes the equipment energy consumption, repair cost associated with component failure and maintenance cost for equipment operation as expressed in equation 5.5:

$$OC = EC + RC + MC \qquad\qquad (5.5)$$

OC = operating cost, expressed in $,
EC = energy cost associated with operating the equipment, in $,
RC = repair cost associated with component failure, in $,
MC = service cost for maintaining equipment operation, in $,

Several primary inputs are needed to evaluate the operating cost such as: the lifetime, discount rate, electricity prices, as well as electricity price trends.

5.3.1 Equipment energy consumption

The equipment energy consumption is the site energy use associated with the use of commercial refrigeration equipment. Although there are potentially some interactive effects on the overall heating and cooling of the building, for purposes of the ANOPR, the LCC analysis includes only the use of electricity by the equipment itself. This approach is consistent with most other DOE equipment efficiency rulemakings. Analysis results from whole building simulation of supermarkets suggest that the overall impact of the design options for the refrigerated cases when taken together did not significantly affect the HVAC energy consumption.

5.3.2 Maintenance costs

The maintenance cost is the cost to the consumer associated with general maintenance, such as checking and maintaining refrigerant charge levels, cleaning heat exchanger coils,... Annualized maintenance costs for commercial refrigeration equipment were taken from DOE reports based on RS Means Facilities Maintenance & Repair Cost Data [DOE07]. RS Means provides estimates on the person-hours, labor rates, and materials required to maintain commercial refrigeration equipment.

Maintenance costs include both preventive activities and lighting maintenance. Preventive maintenance activities for commercial display cases expected to occur on a semi-annual basis as including the following actions: cleaning evaporator coils, drain pans, fans and intake screens; lubricating motors; inspecting door gaskets and seals, and lubricating hinges; cleaning condenser coils; checking refrigerant pressures and compressor oil as necessary; checking starter panels and controls; and checking defrost system operation. However, these activities were not broken into separate line-item maintenance activities since no detailed data were available.

A single figure of $156/yr (in 2008$) for preventive maintenance activities is applied for all commercial refrigeration (DOE value). Moreover, preventive maintenance costs remain constant as equipment efficiency increases since no data were available to indicate how maintenance costs vary with equipment efficiency level.

Lamp replacements and other lighting maintenance activities are considered apart from preventive maintenance and are required for commercial refrigeration equipment. Because the lighting configurations can vary by equipment class and efficiency level, the relative maintenance cost are estimated for each case type and lighting technology. The frequency of failure and replacement of individual lighting components are estimated based on DOE report [DOE07], then an annualized maintenance cost is defined as the sum of the total

lighting maintenance costs (in 2008$) over the estimated life of the equipment divided by the estimated life of the equipment.

Lifetime estimates for particular components were as follows:
• Fluorescent lamps would be replaced every 24 months in a preventive fashion.
• Fluorescent lamp ballasts would be replaced once over the estimated 10-year life of the equipment based on a typical ballast life of 80,000 hours.
• LED lamps would be replaced once over the estimated 10-year life of the equipment based on a typical fixture life of 50,000 hours .

5.3.3 Repair costs

The labor and materials costs associated with repairing or replacing components that have failed. The repair cost is the cost to the consumer for replacing or repairing components in the commercial refrigeration equipment that have failed. The annualized repair cost for baseline energy consumption commercial refrigeration equipment (i.e., the cost the customer pays annually for repairing the equipment) is based on equation 5.6 developed by the DOE:

$$RC = k\frac{EP}{N} \tag{5.6}$$

Where k = fraction of the equipment price (a value of 0.5 was assumed),
EP = equipment price expressed in $,
N = average lifetime of the equipment in years (a value of 10.0 years was assumed).

Since no data were available to indicate how repair costs vary with equipment efficiency level, they were taken constant.

5.3.4 Lifetime

Lifetime t expresses the age at which the commercial refrigeration equipment is retired from service. A typical lifetime of 10 years is appropriate for commercial refrigeration equipment based on [DOE07].and discussion with experts.

5.3.5 Discount rate

The discount rate "r" expresses the rate at which future costs are discounted to establish their present value. The discount rate varies accordingly with economic sectors and store categories. Based on [DOE07], a discount rate of 4.76% is considered after deducting expected inflation from the cost of capital.

5.3.6 Electricity prices

Electricity prices used in the analysis are the price per kilowatt-hour in cents or dollars (e.g., cents/kWh) paid by each customer for electricity. Because of the wide variation in electricity consumption patterns, wholesale costs, and retail rates across the US, regional differences in electricity prices were considered .Electricity prices are determined using average commercial electricity prices in each State, as determined from Energy Information and used by the Department of Energy of the US government. Table 5.2 provides data on the adjusted electricity prices for different states.

Table 5. 1 Commercial electricity prices cents/kWh ([DOE07])

State	Commercial Electricity Price	State	Commercial Electricity Price	State	Commercial Electricity Price
Alabama	7.32	Kentucky	5.73	North Dakota	6.02
Alaska	11.20	Louisiana	7.92	Ohio	8.06
Arizona	7.57	Maine	11.04	Oklahoma	6.82
Arkansas	5.91	Maryland	7.42	Oregon	6.81
California	13.01	Massachusetts	11.18	Pennsylvania	9.20
Colorado	7.05	Michigan	8.06	Rhode Island	10.77
Connecticut	10.60	Minnesota	6.54	South Carolina	7.27
Delaware	7.81	Mississippi	7.74	South Dakota	6.44
Dist. of Col.	7.85	Missouri	6.17	Tennessee	7.13
Florida	7.62	Montana	7.31	Texas	8.37
Georgia	7.11	Nebraska	6.20	Utah	5.96
Hawaii	16.04	Nevada	9.38	Vermont	12.05
Idaho	5.94	New Hampshire	10.99	Virginia	6.13
Illinois	7.79	New Jersey	9.72	Washington	6.48
Indiana	6.54	New Mexico	7.85	West Virginia	5.82
Iowa	6.67	New York	13.80	Wisconsin	7.44
Kansas	6.85	North Carolina	7.10	Wyoming	6.13

Furthermore, DOE recognized that different kinds of businesses typically use electricity in different amounts at different times of the day, week, and year, and therefore face different effective prices. To make this adjustment, average prices paid by the four kinds of businesses were identified and considered in this analysis compared with the average prices paid by all commercial customers.

$$Eprice_{Bld,CA} = Eprice_{CA} \times \frac{Eprice_{Bld,USA}}{Eprice_{USA}}$$ (5.7)

Where $Eprice_{Bld,CA}$ = average commercial sector electricity price in a specific building in California in year 2008,
$Eprice_{CA}$ = average commercial sector electricity price in California in year 2008,
$Eprice_{Bld,USA}$ = national average commercial sector electricity price in the considered building,
$Eprice_{USA}$ = national average commercial sector electricity price.

Table 5.3 shows the derivation of electricity price ratios for different businesses/building types.

Table 5. 2 Electricity price ratios for different businesses ([DOE07].)

Business type	Grocery/ Store food	Convenience store	Convenience store with gas station	Other	All food sales	All Commercial buildings
Electricity Price (cents/kWh)	7.2	8.6	7.7	8.2	7.6	7.8
Ratio of electricity price to average price for commercial buildings	0.92	1.10	0.99	1.05	0.97	1.00

5.3.7 Electricity price trends

The electricity price trend provides the relative change in electricity prices for future year out to year 2017 corresponding to a lifetime of 10 years considered for this study. The EIA's Annual Energy Outlook AEO 2006 reference case is applied to forecast future electricity prices for the LCC analysis presented in this work. Figure 5.1 illustrates the electricity price trend.
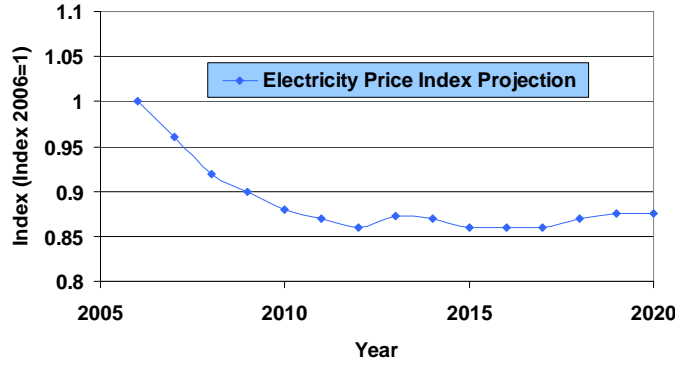
Figure 5. 1Electricity price trend out to year 2020 ([DOE07].)

5.4   Pay Back Period (PBP)

The PBP (Pay Back Period) is the change in purchase cost due to an increased efficiency standard divided by the change in annual operating cost that results from the standard. It represents the number of years it will take the customer to recover the increased purchase cost through decreased operating costs. In the calculation of PBP, future costs are not discounted. Inputs to PBP analysis are categorized as presented for the LCC analysis, i.e. inputs for establishing the total installed cost and inputs for calculating the operating cost

Numerically, the PBP is the ratio of the increase in purchase cost (i.e., from a less efficient design to a more efficient design) to the decrease in annual operating expenditures. The equation 5.8 shows PBP expression:

$$PBP = \frac{\ln\left(\dfrac{\Delta OC}{\Delta IC}\right) - \ln\left(\dfrac{\Delta OC}{\Delta IC} - \dfrac{r - apel}{1 + apel}\right)}{\ln\left(\dfrac{1 + r}{1 + apel}\right)} \qquad (5.8)$$

where PBP = payback period in years,
ΔIC = difference in the total installed cost between a more efficient equipment and baseline equipment,
ΔOC = difference in annual operating costs,
r = discount rate,
and $apel$ = actualization rate of the electricity price.

Payback periods are expressed in years. Payback periods greater than the life of the product mean that the increased total installed cost of the more efficient equipment is not recovered in reduced operating costs over the life of the equipment. Hence, The PBP can be computed only when the following condition is fulfilled (equation 5.9):

$$\frac{\Delta OC}{\Delta IC} = \frac{OC_{Base} - OC_{Efficient}}{IC_{Efficient} - IC_{Base}} > \frac{r - apel}{1 + apel} \qquad (5.9)$$

In the present work, two LCC analysis are conducted. The first one aims at defining the optimal aggregation of technical options for energy savings in a supermarket. The second analysis establishes the distribution of LCC differences between the baseline refrigerating system (direct expansion) and other systems (distributed and secondary loop systems).

5.5   Life cycle cost assessment (LCCA) of the technical options in an aggregated model.

This section presents LCC results for higher efficiency and energy saving options specified in the previous intermediate report. A screening analysis was conducted in order to chose the technologies to be evaluated, and implemented as design options in the energy consumption model. The investigated design options are:
- Higher efficiency lighting and ballasts for equipment families (LED lighting),
- Closing open display cases with glass doors,
- Higher efficiency evaporator fan motors (ECM motors),
- Defrost cycle control,
- Anti-sweat heater control,
- Installing night shields for open display cases.

First, baseline case LCC and each technical option LCC are calculated separately, and then aggregated models of technical options are studied. For a given option, LCC is calculated according to the methodology presented in section 1. Hence, the impact of applying a technical option to a baseline equipment is evaluated in terms of total installed cost and operating cost.

5.5.1 Baseline scenario

Before evaluating incremental cost induced by applying energy saving options, the baseline supermarket LCC must be evaluated. DOE provided in its Technical Support Document for commercial refrigeration the baseline equipment price, maintenance, repair and installation costs ([DOE07].). Therefore, the Commercial Refrigeration Equipment Families defined in the intermediate report were compared to the families defined in the DOE document in order to define corresponding cost and prices.

Table 5.4 illustrates commercial refrigeration display case families and detailed costs used to establish the LCC analysis.

Table 5. 3 Display cases categories and corresponding installed and operating costs[DOE07]

DESCRIPTION			INSTALLED COST				OPERATING COST						
Position	Family	Tº level	MSP ($)	EP ($)	InstC ($)	IC ($)	RC ($)	PM ($)	LM ($)	MC ($)	EC ($/2008)	OC ($)	Symbol
Vertical	Open	Medium	3944	5478	365	6118	274	156	132	288	1807	2369	DC-1/ DC-10
Vertical	Open	Low	6636	9217	365	9857	461	156	43	199	4671	5331	DC-6
Vertical	Closed	Medium	6546	9092	365	9742	455	156	70	226	893	1574	DC-2 / DC-14
Vertical	Closed	Low	6664	9256.	365	9906	463	156	96	252	1964	2679	DC-7
Semi-Vertical	Open	Medium	3890	5403	365	6043	270	156	88	244	1307	1821	DC-3
Service Over Counter	Closed	Medium	7960	11056	365	11696	553	156	73	229	1240	2022	DC-12
Horizontal	Open	Medium	3922	5448	365	6087	272	156	0	156	254	683	DC-5 / DC-9
Horizontal	Open	Low	4134	5742	365	6382	287	156	0	156	1375	1818	DC-13

EC = energy cost associated with operating the equipment,	EP = equipment price
IC= total installed cost,	InstC= installation cost
LM = lighting maintenance.	MC = service cost for maintaining equipment operation
MSP = manufacturer selling price,	OC = operating cost,
PM= preventative maintenance costs,	RC = repair cost associated with component failure,

5.5.2 Energy saving technologies

A number of technologies that could potentially be used to improve the efficiency of commercial refrigeration equipment was considered to evaluate energy savings in commercial refrigeration. These include higher efficiency lighting, higher efficiency fan motors, defrost cycle control, anti sweat heater control, and door installation for open display cases (except for display cases dedicated for vegetables and fruits).

Higher efficiency lighting (LED)

Higher efficiency lighting leads to energy savings in two ways: less energy is used directly for lighting, and less heat energy is dissipated into the refrigerated case by the lamp. The most recent trend in case lighting is the use of light emitting diode (LED) technology that allow comparable product illumination with less total wattage. Therefore, LED technology will be considered to evaluate energy savings from lighting.

Higher efficiency evaporator fan motors (ECM)

The electronically commutated permanent magnet motor (ECM), a three phase electric motor, is more energy efficient than either shaded pole or PSC motors but ECM motors are more expensive than equivalent PSC motors. ECM motors are regarded in this report as an energy saving technical option, and energy savings induced are evaluated.

Anti sweat heaters controllers (ASH)

Anti sweat heating is necessary to prevent moisture condensation on surfaces of display cases , which temperature can drop below ambient dew point. Anti sweat heating controllers match the on-time of the anti-sweat heaters to the anti-sweat heating requirements imposed by the ambient humidity, reducing energy consumption when the ambient humidity is low.

Defrost Cycle Control

As the air in the refrigerated space is cooled, water vapor condenses on the evaporator coil surface. In refrigerators and freezers, where evaporator coil is below 32°F, this water freezes, forming a growing frost layer, increasing the thermal resistance to heat transfer from the coil to the air, reducing thus the cooling performance. Among available defrosting mechanisms, electric defrost is investigated in this study. It involves melting frost by briefly turning on an electric resistance heater, near or in contact with the evaporator coil. However, for energy saving purposes, defrost cycle control is needed to minimize energy required for defrosting. Defrost cycle control considered in this report involves management of the initiation and termination of defrost cycles, and thereby the frequency and duration of defrosting cycles according to frost conditions determined by temperature sensors.

Door installation for open display cases

Refrigerated display cases without doors allow consumers to have easy access to products while maintaining temperatures that ensure food safety. The refrigeration load of such cases is dominated by entrainment of warm and moist air into the case (called infiltration). Reduction in total case energy consumption can be achieved by installing doors for open display cases when it is possible, in order to reduce the infiltration load as well as the induced frost formation on the evaporator coil.

5.5.3 Total installed cost

The total installed cost equals the customer equipment price plus the installation price. Therefore, implementing a new option may incur incremental costs on either equipment price, installation cost or on both. These cost increases are based on data taken from technical literature. Table 5.5 illustrates for each energy saving options, the corresponding incremental cost to be added to equipment price and installation cost as well as the reference where these values are taken from. Blank cells mean no incremental cost is incurred.

It should be noted that installation cost does not taken into account the cost of replacing baseline options by energy saving ones.

Table 5. 4 Impact of energy saving options on total installed cost

Technical Options	Equipment price	Installation cost	Reference
LED Lighting	Increase of 53$		[CCR08], [SMA08]
ECM motors	Increase of 50$		[ACE04]
ASH control	Increase of 14$/ft of cabinet length		[PGE07]
Doors Installation	Replace by equivalent equipment with doors		Results of Simulations
Night Shield	Increase of 204$/m of cabinet length		[PGE07]
Defrost Control	Increase of 14$/ft of cabinet length		[CCR08]

5.5.4 Total operating cost

The operating expenses include repair and maintenance cost as well as energy consumption cost. The energy consumption of a commercial refrigerating equipment is based on the model developed and presented in section 2. Table 5.6 shows the impact of applying energy saving options on maintenance, repair and energy costs as well as references where these values are taken from.

Table 5. 5 Impact of implementing higher efficiency options on operating cost

Technical option	Baseline case	Repair Cost	Preventive Maintenance Cost	Lighting maintenance	Energy Cost
LED Lighting	T8 linear fluorescent lighting	Lower repair frequency		Lower maintenance frequency	Lower lighting power
ECM motors	Brushless DC motors				Lower fan consumption
ASH control	No ASH control				Lower energy consumption
Doors Installation	Left open cases				Modify supermarket layout
Night Shield	NO night shield				Lower energy consumption
Defrost Control	NO defrost control				Lower energy consumption

5.5.5 Centralized system energy consumption

Table 5. 6 Energy consumption evaluation for possible energy saving options

DC description	Baseline W/mL	Baseline W/m^3	LED	ECM	ASH Control	Defrost Control	Doors Installation	Combined Options
VOPMT	905	464	7%	3%	0%	1%	44%	57%
SVOPMT	662	445	9%	5%	0%	1%	47%	65%
HOPMT	410	653	14%	7%	0%	2%	10%	34%
VOPLT	2435	1015	2%	1%	3%	0%	75%	80%
SVOPLT	1800	1071	3%	2%	4%	0%	73%	80%
HOPLT	801	598	7%	4%	1%	1%	56%	68%
VGDMT	411	194	14%	7%	9%	0%	0%	30%
SVGDMT	385	283	15%	8%	9%	0%	0%	29%
VGDLT	603	142	10%	5%	6%	0%	0%	18%
HGDLT	342	380	17%	9%	3%	0%	0%	26%

Table 5. 7 Break down of a supermarket energy consumption due to centralized refrigeration system

SUPERMARKET	Centralized System Energy Consumption						
DC description	Baseline	LED	ECM	ASH	DEF	DOORS	Combined
DC Supermarket Consumption (kWh)	166	140	153	158	164	143	92
DC Energy Consumption reduction (%)	0%	15%	8%	5%	1%	14%	44%

Where   V: vertical   H: horizontal   SV: semi-vertical   OP: open   GD: glass door   LT: low temperature   MT: medium temperature

The thermal load of display cases found in a typical supermarket is calculated using the analytical model presented in section 2 The results for the baseline are summarized in table 5.7 in terms of refrigeration load per meter of display case (second column), then per m^3 of refrigerated volume (third column).Tables 5.7 summarizes the relative energy gains for each type of refrigerated display case and taking into account the improvement of each technical option and then combining the options where it can be seen that the integration of all option is different than the sum of each. For example, an energy consumption reduction of 57% can be achieved for Vertical open medium temperature case (VOPMT) when combining all options (Table 5.7).

For a baseline scenario, an hourly energy consumption of 166 kWh is calculated. By combining all of the presented options and taking into account the type of display cases defined for a typical Californian supermarket, the energy consumption can be reduced of 44% (Table 5.8).

The effect of door installation is highlighted in figure 5.2 where the contribution of energy saving option is plotted, for both vertical and horizontal open display cases, medium and low temperature equipments.
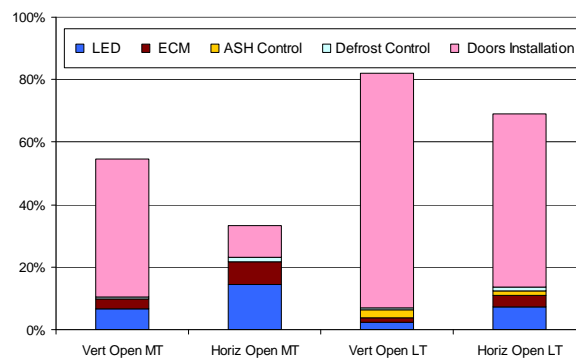
Figure 5. 2 Energy saving option contribution to equipment energy consumption

Figure 5.3 shows the energy saving breakdown for closed display cases. It appears that the refrigeration load can be reduced by approximately 30% for these display cases. Most of this energy saving is due to LED lighting systems as one can see on figure 5.3.
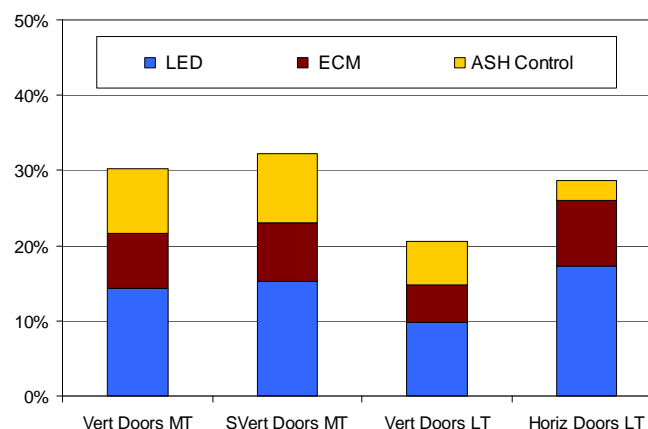
Figure 5. 3 Energy savings breakdown for closed display cases

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5.5.6 Results of LCC analysis

The LCC analysis is conducted for a typical supermarket with a lay-out presented in section 1.5. Assuming a lifetime of 10 years for all commercial refrigerating equipments of this layout, the LCC of a supermarket is obtained by summing the LCC of each refrigerating equipment.

After defining initial prices, costs and possible energy savings for each commercial refrigerating equipment found in the lay-out, the equipment LCC and corresponding PBP are evaluated according to equation [1] and equation [7] respectively and presented in table 5.9.

Table 5. 8 LCC and PBP of investigated technical options for a typical supermarket layout

Commercial Refrigeration Equipment Options	PBP (years)	LCC ($)
Baseline	0	3,130,402
LED Lighting	0.3	2,923,013
ECM Fan Motors	0.5	3,039,767
ASH control	1.8	3,081,110
Installing Doors	3.2	3,010,195
Defrost Control	3.7	3,117,375

Once the options are evaluated separately, the LCC of aggregated options are estimated. For the aggregated models, technical options are successively implemented according to an increasing PBP. Figure 5.4 shows the LCC distribution for different tested aggregated models. It appears that applying LED lighting, ECM fan motors and installing doors, controlling anti-sweat heater and defrost mechanisms are profitable since a decreasing tendency is observed and the lowest LCC is calculated.
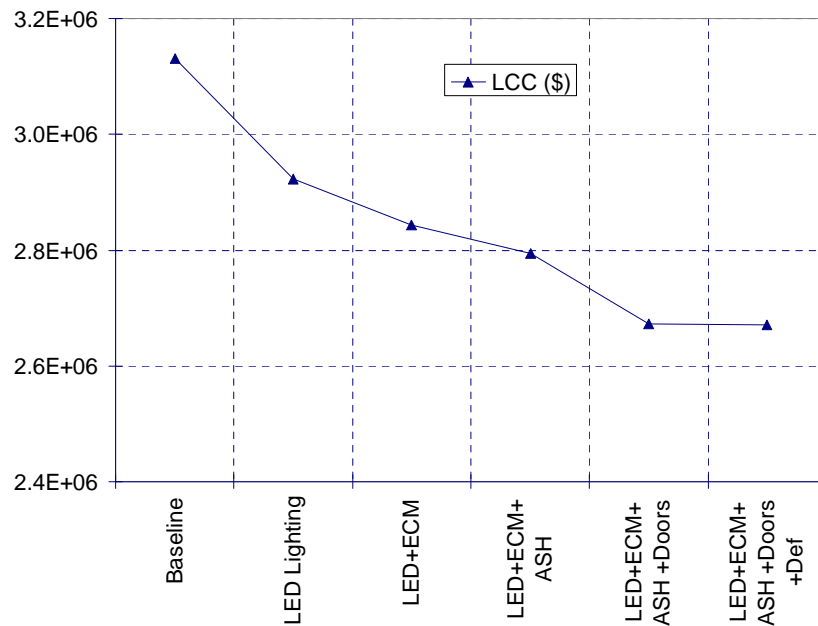
Figure 5. 4 LCCA of a supermarket over a 10 year lifetime

By applying all of the investigated energy saving options, the complete aggregated model allowed 17% savings on the LCC when compared to the baseline scenario.

5.6 Analysis of direct and indirect systems in terms of energy efficiency and costs

A second LCC analysis is dedicated to define the most appropriate refrigerating system in terms of energy efficiency and costs. To begin with, a state of the art is conducted in order to screen existing refrigeration systems and choose systems to be investigated in the LCCA.

Typical supermarket refrigeration systems consist of direct expansion air/refrigerant coils located in display cases and walk-in coolers. Compressors are located in a machine room, in a remote part of the refrigerated store, either in the back area or on the roof. Condensers are located either in the machine room, or more likely, on the roof above the machine room. Piping is connecting back and forth between the machine room and the refrigerating equipment for refrigerant circulation either in liquid phase or in vapor phase (see figure 5.6).

The difference between a secondary loop and a direct expansion refrigeration system is that the refrigeration of display cases and walk-in coolers is provided by a chilled secondary fluid called Heat transfer fluid (HTF), pumped from a primary heat exchanger in the machinery room where the refrigerant evaporates and cools the HTF to the display cases (see figure 5.9).

The most commonly used secondary fluid in both commercial and industrial applications is mono-propylene glycol (MPG) for medium temperature racks but since 10 years a number a products have been proposed based for example on acetate formiate. CO_2 is a promising heat transfer fluid for low temperature units. These two HTFs will be used thereafter to evaluate secondary loop performances and to carry out a LCC analysis.

Five refrigeration systems are investigated in the second LCC analysis. In addition to multiplex direct expansion system and distributed one, three secondary loop refrigeration systems are described. The comparative study is conducted for a typical supermarket of 4,400 m^2 (47,360 ft^2).

The first step of the analysis consists of evaluating the refrigeration load of a supermarket. The input power required by compressors for those load conditions is then calculated for each temperature bin (low and medium temperature racks) and the number of operating compressors is inferred.

The typical Californian supermarket includes two low temperature racks and two medium temperature racks. Display cases and coolers are grouped and connected to compressors racks according to the required saturated suction temperature (SST) to maintain the desired case temperature. In the following analysis, low temperature racks will operate at - 32°C (-25°F) SST whereas medium temperature racks will operate at -10°C (+14°F) SST. Each compressor rack consists of three or four compressors sized in a way that allows compressors, operating simultaneously, to provide the cooling capacity that meets the design refrigeration load.

For the 47,360 ft^2 supermarket, the refrigeration capacity is190 kW at the medium temperature level and about 150 kW at the low temperature level. Assuming four compressors in each rack, the refrigerating capacity of a compressor operating at the medium temperature level is 32 kW whereas a cooling capacity of 19 kW corresponds to compressors mounted on low temperature racks.

The most common type of condensers used in supermarket refrigeration is air-cooled condensers. These condensers usually employ finned-coil construction with 8 -10 fins/inch (300-400 fins/m) and multiple fans (figure 5.5). Air cooled condensers are known to operate reliably and require the least maintenance.

Figure 5. 5 Roof top air cooled condenser

5.6.1 Multiplex direct expansion refrigeration System

System description

The most common new direct system in supermarkets is the multiplex refrigeration system using R-404A as a working fluid. It consists of multiple racks of compressors operating at the same saturated suction temperature with common suction and discharge refrigeration lines (one for medium temperature and another one for low temperature racks) [BAX03]. The term multiplex refers to the use of multiple compressors piped to a common suction and discharge manifolds, all installed on a skid containing all the necessary piping, control valves and electrical wiring to control the compressors and the refrigeration provided to the display cases.
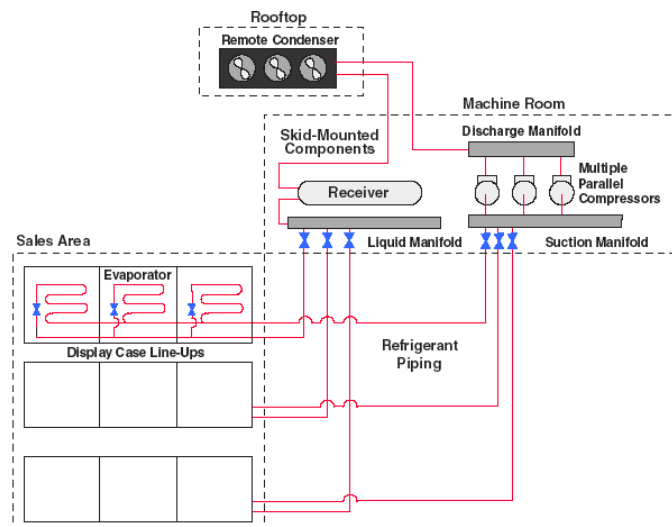
Figure 5. 6 Piping diagram for the Medium Temperature
Multiplex Refrigeration

Figure 5.6 shows elements of a medium temperature unit of a secondary loop refrigeration system. Compressor racks are installed in a machine room with long refrigerant pipes connecting them to display cases in the sales area. The piping length can reach several hundred meters for large supermarkets, implying possible failures, fugitive emissions at joints, and pressure losses especially on the suction line. The hot gas discharge from the compressor is piped back to the remotely located condenser, which condenses gas to liquid. Liquid refrigerant is piped back to the compressor rack, where a receiver, liquid manifold and associated control valves distribute liquid to the cases and walk-ins.

Multiplex refrigeration model

Figure 5.7 shows the diagram of the most commonly used compressor arrangement in a multiplex refrigeration system found in supermarkets. The refrigeration system configuration is described in order to evaluate its performances. Several parameters should be known beforehand such as display case evaporator temperature, minimum condenser temperature, condenser type as well as the refrigerant in use.
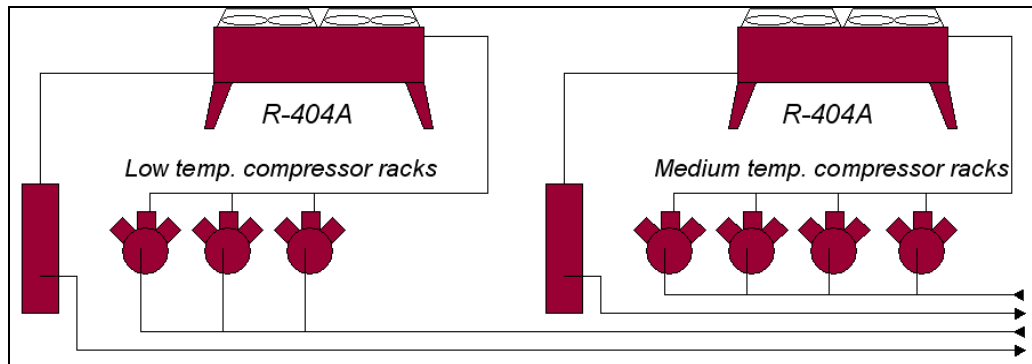
Figure 5. 7 Design of a multiplex refrigeration system using R-404A

Multiplex refrigeration state points

The refrigerant in both temperature units is R-404A. Several operating set points must be monitored for each multiplex compressor rack and for each compressor suction group.
- Saturated Suction Temperature (SST)- the saturation temperature corresponding to the refrigerant pressure measured at the compressor suction.
- Saturated Discharge Temperature (SDT)- the saturation temperature of the refrigerant based upon the pressure measured at the compressor discharge.
- Return Gas Temperature- the refrigerant gas temperature measured at the compressor suction.
- Refrigerant liquid temperature- the liquid temperature measured at the receiver outlet, and before and after sub-cooling heat exchangers if installed.

Once the state points are defined, the cooling capacity and the compressor power can be caculated based on compressor technical sheets supplied by manufacturers (Copeland, Carlyle, Danfoss, Bitzer…).

Operating conditions

The most significant parameter in determining condensing temperature is the temperature difference with the outdoor ambient temperature, ΔT, since heat is rejected to ambient conditions. The condenser ΔT is dependent on the condenser type: For air-cooled condensers considered in this study, 8K and 10K are standard values of ΔT for low and medium temperatures respectively.

The fan power for remote condensers is based upon the condenser type. Air-cooled condensers for low temperature refrigeration are normally sized for a smaller temperature difference ΔT and require more fan power than condensers employed with medium temperature refrigeration [FOS04] [ORL01].

Pressure drop will occur between the cases evaporator and the compressor suction point. This pressure drop is taken into account by a lower saturated temperature value at the compressor suction port. Heat gain to the return gas will also takes place and affects the refrigerant mass flow rate transferred by the compressors. Pressure drops and superheat vary depending on the distance between the display cases and the compressor racks. These factors tend to decrease the capacity of the compressor and increase the run time need to meet the refrigeration load. In this study, a superheat of 10K (18°F) and a pressure drop of 0.3 bar (4 psi abs) are assumed at the compressor suction ports.

The temperature of refrigerant liquid temperature is an important operating state point. This temperature is usually lower than the condensing temperature because the store is air conditioned and so there is a "free" cooling of the refrigerant in the liquid lines. This "free" sub-cooling varies between 10 and 15K. For instance, a refrigerant liquid temperature of 30°C corresponds to a condensing temperature of 45°C.

When a complementary sub-cooling is realized by a complementary small refrigerating which purpose in only to cool the refrigerant in liquid phase , the liquid refrigerant temperature leaving the sub-cooler is typically of 5°C (41°F) this mechanical sub-cooling is not free and its energy consumption in taken into account in the energy consumption calculation. Nevertheless in the present analysis, no mechanical sub-cooling is taken into account and consequently, the liquid refrigerant temperature varies between 20 to 30°C according to condensing temperature related to the climatic conditions.

5.6.2 Distributed system with separate roof condenser

Another option for direct expansion systems is the distributed system with a separate rooftop condenser. Distributed systems may have different designs but the main concept is to install compressors in sound-proof boxes near the display cases, the condensing heat being released on a water circuit connected to dry-air coolers (as shown on figure 5.8).

The sound-proof boxes are located within the store to provide refrigeration to a particular series of display cases, such as meat, dairy, frozen food, etc. With this arrangement, the pressure losses as well a superheat are reduced. The refrigerant suction and liquid lines are shortened in a distributed system and refrigerant charge requirement for the distributed system is reduced compared to a multiplex refrigeration system. The total refrigerant charge will be about 75 % of a direct expansion multiplex system [CAG04].
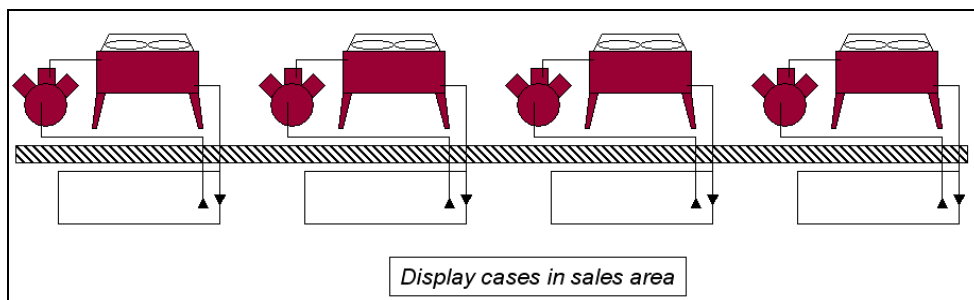
Figure 5. 8 Description of the distributed refrigeration system

5.6.3 Secondary Loop Refrigeration

The secondary loop consists of a HTF pumped between a central chiller and display cases. At least two fluid loops are installed in a supermarket depending upon the refrigeration load

composition and temperature levels. Refrigeration with secondary loop systems has been introduced in supermarkets to decrease the refrigerant charge and to minimize potential refrigerant leakage. Secondary loop systems may have various designs with different energy efficiencies.

The most commonly used HTF is still MPG in the US in both commercial and industrial refrigeration. MPG is preferred because it is inert to common piping materials and most non metallic gaskets and seals. Nevertheless, MPG is used only for medium temperature units: at concentrations needed for low temperature refrigeration, its very high viscosity induces high pumping power. Consequently, other fluids, such as CO_2 are used for low temperature units.

5.6.3.1 Secondary loop system description

Flow rates of single phase HTF are defined by the temperature change of HTF in the whole cooling circuit, it is typically of 4 or 5 K. Because of the high viscosity and the necessary HTF mass flow rate to provide cooling, the energy associated with pumping is substantial. But the picture is significantly different with CO_2 which is now used as a changing phase HTF : CO_2 evaporates partially in the display cases heat exchangers (typically 20% of it), so CO_2 returns at the primary evaporator where the 20% is condensed.

The refrigerant charge in a secondary loop refrigeration system is approximately 50% lower than that of a direct expansion and minimize the pressure drop as well as the superheat on the refrigerant side. All these factors coupled allow on one a reduction in the compressor energy consumption and an increase for the HTF pumping. Figure 5.9 illustrates the piping diagram of medium temperature unit of a secondary loop refrigeration system.
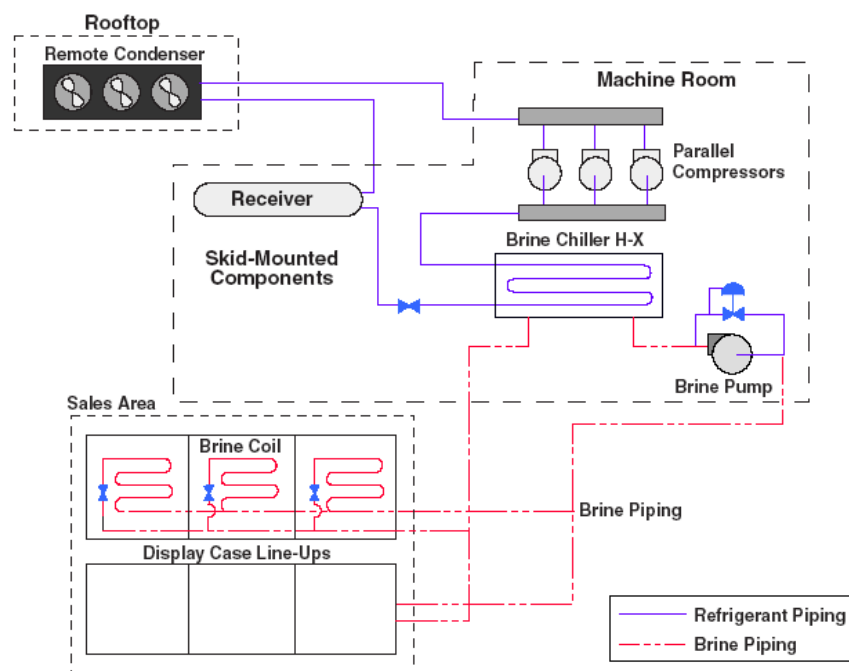
Figure 5. 9 Elements of a secondary loop refrigeration for medium temperature racks

Note : on figure 5.9 brine has to be understood as HTF.

5.6.3.2 Modeling of the Secondary Loop Refrigeration

The major difference in the analysis of a secondary loop refrigeration system is the operation of the secondary loop. The HTF mass flow rate (MFR) is caculated by applying equation 5.10 when MPG is used whereas equation 5.11 is applied for secondary loops using vapor-liquid CO_2.

$$\dot{M}_{SF} = \frac{Q_0}{C_{SF}\Delta T_{SF}} \qquad (5.10)$$

$$\dot{M}_{SF} = \frac{Q_0}{\Delta h_{SF}} \qquad (5.11)$$

Where Q_0 = refrigeration load deliverd by the secondary fluid loop (kW)

$\dot{M}_{SF}$ = HTF mass flow rate (kg/s)

ΔT_{SF} = Temperature change of HTF in the display cases circuit

Δh_{SF} = enthalpy change of HTF in the display case circuit (kJ/kg).

In the secondary loop HTF gains heat not only in the display cases but also heat losses occur in the circuit. The most significant heat gain is due to the operation of the secondary fluid loop pump. The pump power is based on the maximum fluid flow needed to meet the design refrigeration load. The power required by the pump is calculated based on HTF thermo-physical properties at the temperature level, piping length and diameter, pressure drops on all the circuit.

The pump input power is calculated by equation 5.12 assuming its overall efficiency η_p is 55% .

$$W_p = \frac{\dot{M}_{SF}}{\rho_{SF}} H_m \frac{g}{\eta_p} \qquad (5.12)$$

where H_m refers to required pump head expressed in Water Column (WC) and the subscript p stands for pump.

State points of the refrigerating system are determined. The evaporating temperature of the primary evaporator is set at 5 K below the outlet temperature of HTF. The refrigerant superheat is set at 10 K .

The temperature difference between the condensing temperature and outdoor ambient temperature, ΔT, is set 8 K and 10 K for low and medium temperatures respectively, for air-cooled condensers considered in this study.

Pressure drop of the refrigerant vapor to the suction of the compressor is lower than pressure drop in multiplex refrigeration system due to the close coupling of the compressors and heat exchangers. This pressure drop is set at 0.2 bar.

5.6.3.3   Investigated secondary loop refrigeration systems

Secondary loop design with completely indirect refrigeration system

CO_2 is used as a HTF in the low temperature system whereas MPG is used for medium temperature system. The temperature of MPG at the exit of the primary evaporator is -8°C and the return temperature -5°C.

For the MPG loop, the required pump head is set at 23 m WC (Water Column). For CO_2 s the required pump head is set at 15 m WC [FOS04] [ORN01].

Figure 5.10 shows a diagram of the 2 secondary loops and the associated refrigeration systems for medium and low temperatures.
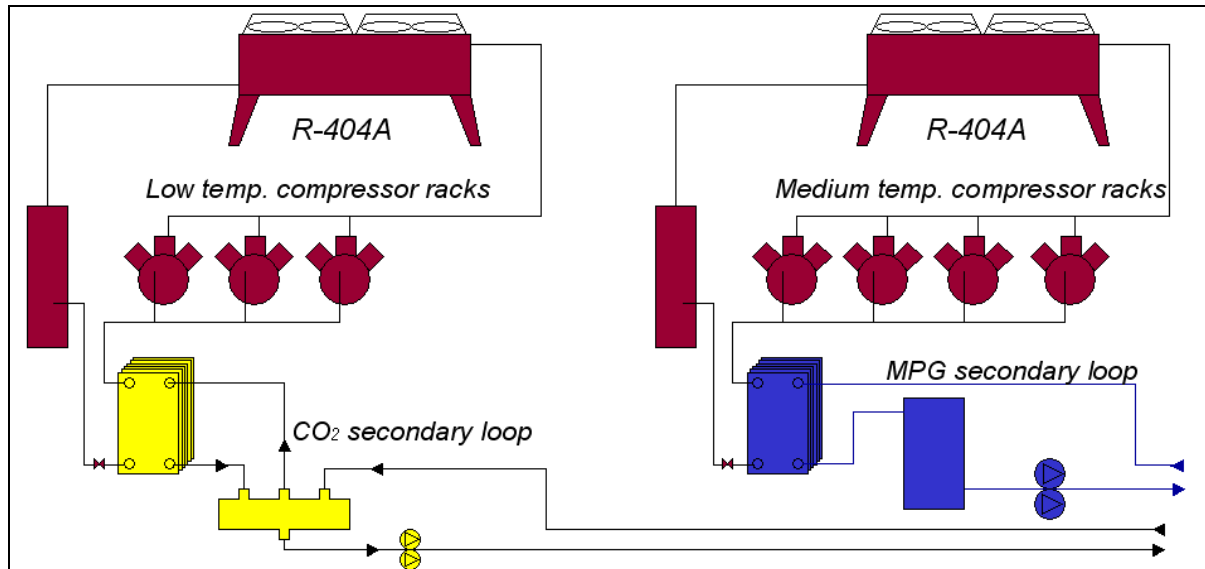
Figure 5. 10 Secondary loop refrigeration system using MPEG and CO_2 for medium and low temperature racks respectively.

Indirect system with CO_2 as the only refrigerant

Another secondary loop systems configuration is studied with CO_2 as the only HTF for both low and medium temperature levels (figure 5.11). CO_2 as a two-phase HTF is not currently used for medium temperature systems. However, some early prototypes exist and such solution might be developed in the near future.
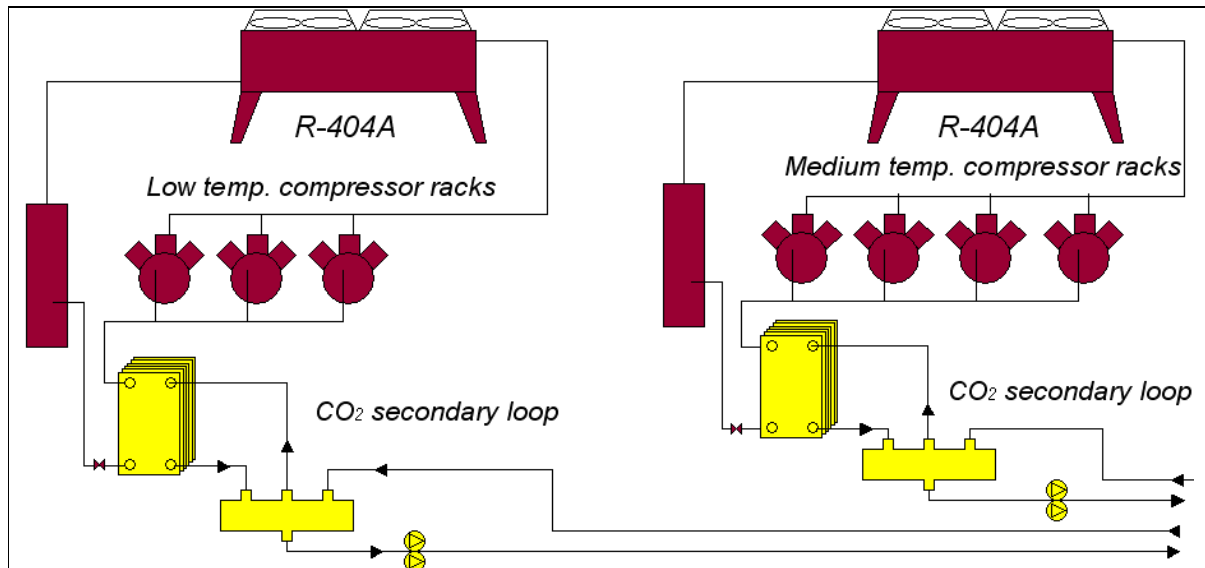
Figure 5. 11 Secondary loop refrigeration system using CO_2 as the only refrigerant

Cascade system with CO$_2$

The cascade system with the CO$_2$ in the low temperature system and a secondary loop using MPG for the medium temperature system is an interesting solution that has been tested in several supermarkets with promising results [CHRIS99]. The HTF at the medium temperature level has a delivery temperature of about -8°C and a return temperature of -5°C.

The delivery temperature of the CO$_2$ in the low temperature unit is about -32°C. Figure 5.12 illustrates the outline of a cascade system with CO$_2$ as secondary fluid for low temperature unit.
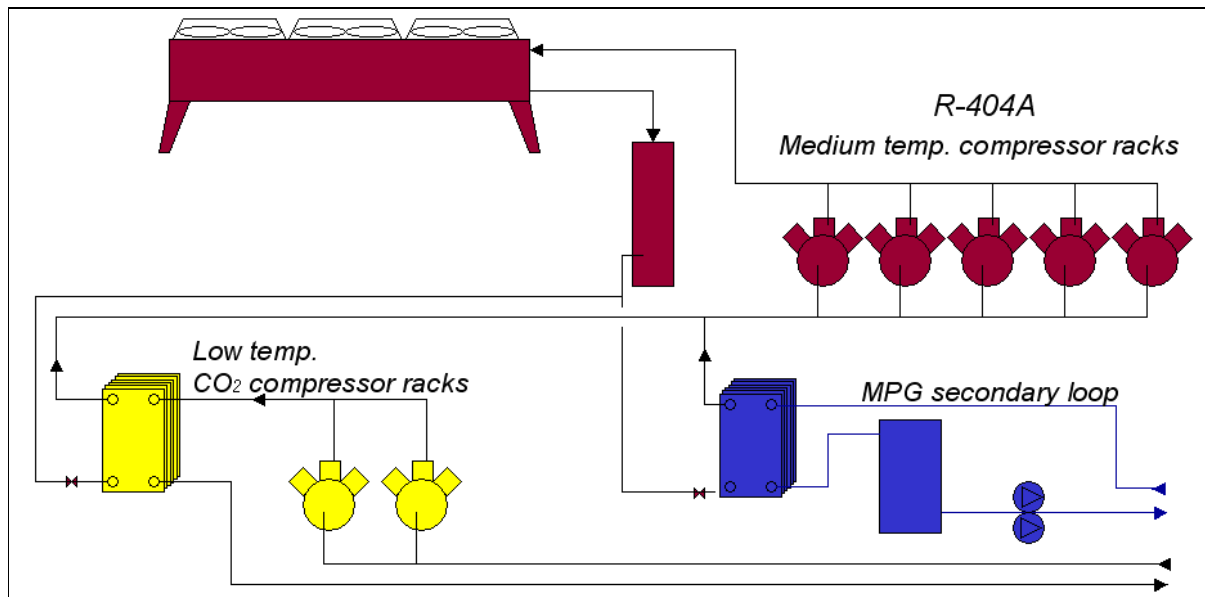
Figure 5. 12 Cascade system with CO$_2$ in the low temperature stage

Table 5.10 summarizes for multiplex refrigeration, refrigerant state points, operating conditions, the chosen compressor, cooling capacity and energy consumption according to the temperature level and refrigerant. Table 5.11 gathers identical information for secondary loop systems investigated in this study, as well as the secondary fluid outgoing and return conditions.

Knowing operating state points and required compressor power, compressors for low and medium temperature racks are chosen from manufacturers catalogs. All compressor models found in tables 5.10 and 5.11 are Copeland technologies except for the CO$_2$ cascade system compressor which is a Bitzer technology.

Important Note : the results of Table 5.11 are based on literature review, and the COPs of the secondary loop systems are very conservative and they do not take into account progresses made in the last 2 years. Based on new data not available before the calculations, energy efficiency of CO$_2$ secondary systems has improved compared to table 5.11 results. So during the review process of this provisional final report, additional calculations will be made to integrate those last progresses.

Table 5. 9 Multiplex refrigeration system operating conditions for two refrigerants R-22 and R-404A.

	Refrig.	T level	SST (°C)	ΔT_0 (K)	SDT (°C)	ΔT_k (K)	SH (K)	ΔP Suction line (K)	T Return Gas (°C)	T Liquid (°C)	ΔP Discharge line (K)	T Suction Comp (°C)	Comp. Model	Cooling Capacity (kW)	Comp. Power input (kW)	COP	Fan Power (kW)	COP_{system}
Multiplex	R22	Med	-10	15	30-45	10	10	3	7	20-30	0.5	-13	D4DA-200X	32.7	12.7	2.57	0.635	2.45
		Low	-32	15	30-45	8	10	5	-17	20-30	0.5	-37	D6TH-2000 $_{SUB}$	19.25	15.5	1.24	0.775	1.18
	R404A	Med	-10	15	30-45	10	10	2	8	20-30	0.5	-12	D3DS-100X	32.5	13.7	2.37	0.685	2.26
		Low	-32	15	30-45	8	10	4	-16	20-30	0.5	-36	D6TH-270X	19.3	15.9	1.21	0.795	1.16

Table 5. 10 Secondary loop refrigeration systems operating conditions with R-404A as a refrigerant

	System Description	Refrig.	Sec. fluid	T level	SST (°C)	ΔT_0 (K)	SDT (°C)	ΔT_0 (K)	SH (K)	ΔP Suction line (K)	T Return Gas (°C)	T Liquid (°C)	ΔP Discharge line (K)	T suction Comp (°C)	T sec fluid (°C)	Comp. Type	Cooling Capacity (kW)	COP	Power Input (kW)			COP_{system}
																			Comp.	Pump	Fan	
Secondary loop	SL indirect MPG-CO2	R404A	MPG	Med	-13	15	30-45	10	10	3	4	20-30	0.5	-16	(-5 -8)	D3DS-150X	30.3	2.28	13.3	1.09	0.665	2.01
		R404A	CO2	Low	-33	15	30-45	8	10	3	-16	20-30	0.5	-36	-28	D6DH-350X	20.7	1.28	16.1	0.09	0.805	1.22
	SL indirect CO2-CO2	R404A	CO2	Med	-13	15	30-45	10	10	3	4	20-30	0.5	-16	-7	D3DS-150X	30.3	2.28	13.3	0.12	0.665	2.15
		R404A	CO2	Low	-33	15	30-45	8	10	3	-16	20-30	0.5	-36	-28	D6DH-350X	20.7	1.28	16.1	0.09	0.805	1.22
	Cascade CO2	R404A	MPG	Med	-13	15	30-45	10	10	2	5	20-30	0	-15	(-5 -8)	D6DL270	55.1	2.04	27	1.09	1.35	1.66
		CO2	-	Low	-32	15	-13	8	10	3	-15	20-30	0.5	-35	-	KP-120-2	19	5	3.8			

5.7   Life cycle cost assessment (LCCA) of screened refrigeration systems

The LCC of the five refrigerating systems described in previous sections are evaluated based on data available in literature and lessons learnt from the field. The methodology used is the same applied previously to define the optimal aggregated technical options, where the total installed cost and operating cost are calculated for the typical Californian supermarket.

5.7.1 Refrigerating system components

Major components of a refrigeration system are: compressors, condensers, evaporators, piping, display cases, walk-ins and miscellaneous electronics including frequency converter, special FI relay AC/DC, construction of safety circuit, compressor control and oil control with PLC.

Therefore, the total installed cost and the operating costs of each of these components will be investigated separately, then summed together to obtain the refrigerating system installed cost and operating cost.

Lifetime and reliability

According to A.D.Little, supermarkets system compressors have a 10-year expected lifetime. Moreover, the typical lifetime of an air-cooled condenser is at most 10 years. Refrigerated display cases are usually replaced for cosmetic reasons prior to the end of their life and replacement occurs at 5-15 years, depending on store policy. Therefore, the systems are expected to operate reliably for 10 years if properly installed and maintained [LIT96]. For the LCC estimation, the lifetime of 10 years is assumed for different refrigeration systems components.

5.7.2 Conventional direct expansion refrigeration system

The multiplex system with air-cooled condensing is considered the baseline, since it is the most commonly installed configuration now used in supermarkets.

Total installed Cost

The total installed cost of a refrigeration system for a typical supermarket varies between 1 million and 1.1 million dollars [LIT96]. Table 5.12 shows the installed cost breakdown based on personal communication with supermarket industry representatives held by A.D.Little.

Table 5. 11 Installed cost breakdown for a typical supermarket refrigeration system

REFRIGERATION SYSTEM COMPONENTS	INSTALLED COST SHARE
Compressors	18.0%
Walk in evaporators	3.0%
Condensers	5.0%
Miscellaneous electronics	6.0%
Piping	2.5%
Display cases	56.5%
Walk in	9.0%

Based on A.D.Little report on commercial refrigeration systems, the installed cost of system components are evaluated based on the breakdown of installed cost presented. Moreover, in the countries final reports to IAE Annex 26, UK provided the cost of pipework including installation and insulation costs: 50,000$ for direct and 80,000$ for secondary loop systems [UK03].

Table 5. 12 Typical supermarket refrigeration system cost and energy consumption breakdown

DX Components	Component Installed Cost($)	Component price ($)	Component Installation cost ($)	Energy consumption (kWh/year) -
Compressors	195510	147000	48510	860000
Evaporators	47460	21000	26460	0
Condensers	64050	42000	22050	99300
Miscellaneous electronics	92190	52500	39690	0
Piping	67305	21000	46305	0
Display Cases	505575	472500	33075	477800
Walk in	77910	73500	4410	133800
Total	1050000	829500	220500	1570900

Maintenance Cost

Costs for refrigeration system maintenance are roughly 0.25% of supermarket revenues. The maintenance cost for a multiplex refrigeration system is about 75$ per 100 sq.ft.of store sales area, which gives a maintenance cost of approximately 20,000$ for a typical supermarket of 27,000 sq ft [LIT96].

5.7.3 Investigated refrigeration systems

The investigation are based on literature reviews, experts opinions and personal communication with supermarket industry representatives. These investigations considered the total cost of the system including cases, piping, refrigerant, brine, and labor in addition to the compressor rack or primary chiller with the exception of the condenser subsystem. It was considered identical for all of the refrigeration systems, hence no cost premium is incurred when comparing to the baseline refrigeration system.

Distributed systems

Predicted energy consumption savings for a distributed system compared to a conventional multiplex refrigeration system is 12% [BAX03]. The estimated installed cost premiums for distributed are presented in [ORL01]. Estimates are based on actual construction budgets supplied by engineering departments of visited supermarket chains. The distributed system shows higher equipment cost when compared to conventional direct expansion system with an incremental equipment price of 53,000$. However, only a small increase in installation cost is observed and an incremental cost of 7,000$ is estimated. This can be attributed to reduced refrigeration piping cost, but also increased electrical and fluid loop costs.

Secondary loop systems

In secondary loop systems, incrementally higher costs would be incurred in several areas: additional hardware costs of the secondary loop fluid circulation pumps, fluids reservoirs, the secondary fluid itself, the refrigerant evaporator to chill the secondary fluid. The incremental cost

is estimated to 50,000$ for a typical supermarket. Different areas primarily influence the total additional charge [CHRIS99]:

1. the electrical board especially influenced by the frequency converter and compressor control, and the safety circuit construction due to flammability. A.D.Little estimated an additional cost of 10,000$ to account for alarms and emergency ventilation of the mechanical equipment room [LIT02]
2. the assembly and the construction of the refrigeration system

In J. Arias thesis, the investment cost for the direct system is assumed to be 10% cheaper than that of an indirect system according to Bjerkhög, who is responsible for the implementation of a new refrigeration syste0m in the supermarket chain COOP Sweden (part of Kooperation) [ARIA05]. On the other hand, [ORL01].estimated equipment and installation cost premiums of a secondary loop system for a typical supermarket based on interviews with supermarket industry professionals. The incremental refrigeration system price is approximately of 70,000$ and the incremental installation cost of 77,000$ Hence a total cost premium of 147,000$ is estimated.

As reported in discussions at the Annex 26 workshop [BAX03], installation cost premiums for secondary loop approaches (using R-404A or R-507A as primary refrigerant and propylene glycol or potassium formiate HTF for secondary loops) were about 15% for typical US markets. It was also noted that maintenance costs for the secondary system should be less.

The Danish country report (Volume 2) [DEN03] compares installation costs and operating efficiencies for a cascade system and R-404A DX systems. A test system installed in a small store (30 kW cooling capacity) was estimated to cost 10 about 20% more than a traditional DX system and to have about the same energy efficiency. With more experience for installers the premium is estimated to drop under 15%. For larger systems, the premium would drop to 10%.

The British country report (Volume 2) announced an increase in overall energy consumption of 30% with the secondary loop refrigeration system. The energy use includes compressor power, condenser fan power, pumping power, and defrost energy. Most of this increase is attributed to pumping energy [UK03]. As for capital costs, the analysis showed the secondary loop system to be approximately 28% more expensive than a conventional direct system. This was confirmed by the UK experience of increased costs between 15% and 30% for secondary systems.

Moreover, for a plant originally constructed with a CO_2 indirect system, smaller pipes could be used for the return and liquid lines, which would compensate for the cost of the additional equipment in the secondary loop and needed safety devices [GIR03]. Figure 5.9 shows the different sizes and insulations of suction and liquid pipelines for different working fluids.
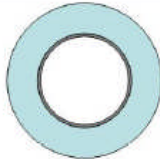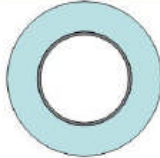
Figure 5. 13 Comparison of liquid and suction pipes for different refrigerating fluids

[CHRIS99] showed that for a cascade system using CO_2 in the low temperature system and a secondary loop using MPG for the medium temperature system, the energy consumption of a secondary loop system decreased by about 5% compared to a conventional supermarket while the investment was 20% higher

Based on these data and on experts opinions, energy savings and cost premiums for secondary loop and distributed systems are estimated. Tables 5.14 through 5.17 illustrates the total installed, equipment price, installation cost as well as energy consumption for each of the 5 investigated refrigeration systems.

In the following tables:
DX: conventional direct expansion system
DIST: distributed system with separate rooftop condenser
SL MPG+ SL CO2-:   secondary loop system with MPG for medium temperature system and CO_2 for the low temperature system
SL CO2 + - :   secondary loop systems with CO_2 as the only refrigerant for both low and medium temperature systems,
Cascade CO2- MPG+ : cascade system with CO_2 in low temperature system and secondary loop with MPG for medium temperature system.

Table 5. 13 Total installed cost for the 5 refrigeration systems components

Refrigeration system Components	DX	DIST	SL MPG+ SL CO2-	SL CO2 + -	CASCADE CO2- MPG+
Compressors	195510	254205	269115	269115	235410
Evaporators	47460	51345	54075	54075	54075
Condensers	64050	92820	64050	64050	64050
Miscellaneous electronics	92190	127050	115395	115395	115395
Pipelines	67305	39165	118650	94920	110355
Display Cases	505575	505890	505575	505575	505575
Walk in	77910	79065	77910	77910	77910
Total Installed Cost	1050000	1149540	1204770	1181040	1162770
Savings% Conventional DX	0%	9%	15%	12%	11%

Table 5. 14 Components prices for the 5 refrigeration systems

Refrigeration system Components	DX	DIST	SL MPG+ SL CO2-	SL CO2 + -	CASCADE CO2- MPG+
Compressors	147000	199080	207375	207375	182490
evaporators	21000	24885	21000	21000	21000
condensers	42000	66360	42000	42000	42000
Misc electronics	52500	82950	58065	58065	58065
Pipelines	21000	8295	41475	33180	33180
Display Cases	472500	472815	472500	472500	472500
Walk in	73500	74655	73500	73500	73500
Total equipment cost	829500	929040	915915	907620	882735
Increase% Conventional DX	0%	12%	10%	9%	6%

Table 5. 15 Components installation cost for the 5 refrigeration systems

Refrigeration system Components	DX	DIST	SL MPG+ SL CO2-	SL CO2 + -	CASCADE CO2- MPG+
Compressors	48510	55125	61740	61740	52920
evaporators	26460	26460	33075	33075	33075
condensers	22050	26460	22050	22050	22050
Misc electronics	39690	44100	57330	57330	57330
Pipelines	46305	30870	77175	61740	77175
Display Cases	33075	33075	33075	33075	33075
Walk in	4410	4410	4410	4410	4410
Total installation cost	220500	220500	288855	273420	280035
Increase% Conventional DX	0%	0%	31%	24%	27%

Table 5. 16 Components energy consumption for the 5 refrigeration systems

Refrigeration system Components	DX	DIST (rooftop design)	SL MPG+ SL CO2-	SL CO2 + -	CASCADE CO2- MPG+
Compressors	55%	50%	66%	60%	54%
evaporators	0%	0%	0%	0%	0%
condensers	6%	6%	6%	6%	6%
Misc electronics	0%	0%	0%	0%	0%
Pipelines	0%	0%	0%	0%	0%
Display Cases	30%	30%	30%	30%	30%
Walk in	9%	9%	9%	9%	9%
Total energy consumption (kWh/year)	1,570,900	1,492,355	1,743,699	1,649,445	1,555,191
Savings% Conventional DX	0%	-5%	+11%	+5%	-1%

5.7.4 LCC analysis results for refrigeration systems

Figure 5.14 illustrates results of LCC from simulations of direct and indirect systems. The period of study was 10 years, the interest rate was 4%, the annual price increase of electricity was 1%.
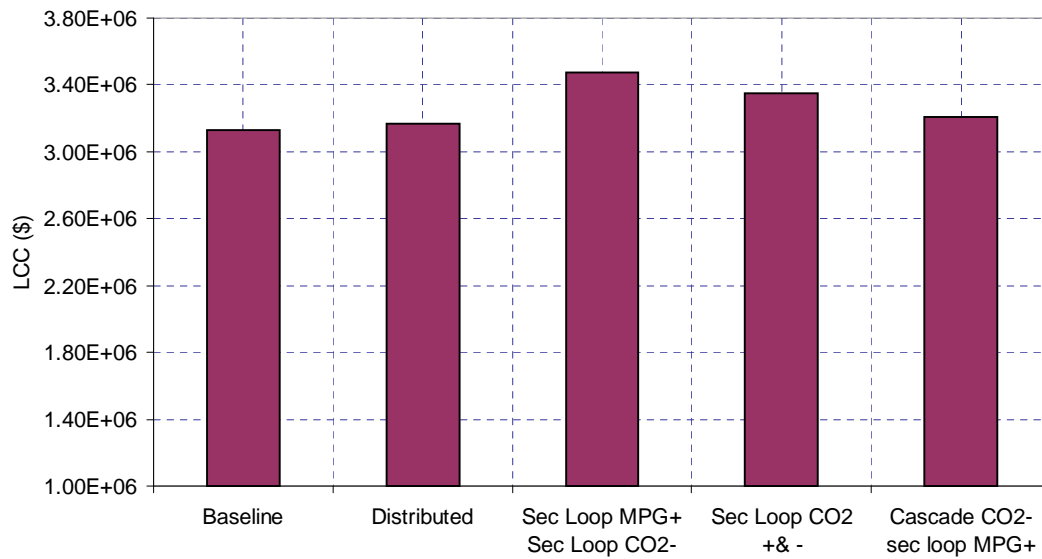
Figure 5. 14: LCC distribution for investigated refrigerating systems

Simulations results and capital costs analyses show that for indirect systems to become attractive alternatives to direct systems, design improvements need to be implemented to reduce both capital and running costs of secondary loop systems. More stringent legislation and incentives may also contribute to a wider application of secondary loop systems. This can be seen by comparing the LCC of secondary loop and distributed systems to the Direct expansion system LCC (figure 5.14). These results are identical to results proposed in the British report to IAE Annex 26 [UK03].

A LCCP of indirect systems and distributed systems is essential to evaluate CO_2 emissions due to the refrigerating system operation (indirect emissions) and the direct emissions of refrigerants taking into account their GWP (global warming potential). The results of LCCP will underline the advantages of secondary loop systems in comparison with the base line (current centralized direct expansion systems).

6. TEWI analysis of refrigeration systems

The basic concept of both the Total Equivalent Warming Impact (TEWI) and the Life Cycle Climate Performance (LCCP) is, for a given product or activity, to rigorously identify all of the warming impacts due to the use of the product through its lifetime. The two major contributors to emissions of greenhouse gases during the lifetime of refrigerating systems in supermarkets are the "indirect" effect of carbon dioxide emissions related to the energy consumption of the product during operation (indirect emissions) and the "direct" effect of greenhouse emissions from the product taking into account their GWP.

Therefore, the contribution to global warming (TEWI) of any refrigerating systems has to be evaluated taking into account both the energy consumption and the refrigerant emissions. Five technologies of refrigeration systems have been compared:
- the base line is the direct expansion system
- distributed system
- secondary loop system (MPG for medium temperature display cases, and CO_2 for low temperature display cases.
- secondary loop system (CO_2 for both medium and low temperature display cases)
- CO_2 cascade and secondary loop (MPG) at medium temperature.

The total equivalent Warming Impact is expressed in equation 6.1:

$$TEWI = GWP \times \lfloor Ln + m(1 - \alpha) \rfloor + nEB \qquad (6.1)$$

with
GWP	Global Warming Potential (kg eq. CO_2)
L	Leakage rate (kg / year)
n	Operation life of the system (years)
m	Fluid quantity charged in the system (kg)
a	Recycling rate (kg of recovered fluid/initial charge)
E	Annual energy consumption (kWh / year)
b	CO_2 emission per electric kWh of produced power (kgCO_2 / kWh)
TEWI	Total Equivalent Warming Impact (kg of CO_2 produced during the lifespan of the equipment).

The direct contribution, due to refrigerant emissions during the life time of the system is expressed as in equation 6.2.

$$Dc = M \times \lfloor a + rc(1 - fre) + ne + s(1 - sre) \rfloor GWP \qquad (6.2)$$

with:
Dc	Direct contribution (kg CO_2 eq.)
M	Nominal charge of refrigerant in the system (kg)
a	Initial charge emission rate (%)
rc	remaining charge (%) at end of life, before decomissioning
fre	Recovery efficiency at decomissioning (%)
n	Operation life of the system (years)
e	fugitive emission rate, including losses due to rupture, and annual maintenance (%)
s	number of renewing operations asking for a complete refrigerant recovery (retrofit for example), except end of life recovery.
sre	refrigerant recovery efficiency when renewing operation are conducted
GWP	Global Warming Impact (kg equivalent CO_2)

Large emissions due to tube or component ruptures have been considered in the fugitive emission rate, which is and average value for a wide number of installations. Servicing and maintenance operations contribute to additional refrigerant emissions depending on the operation quality. This contribution is also included in the fugitive emission rate. Refrigerant losses occurred at the end of life of the system, after decommissioning, when recovery is not appropriately done. A recovery efficiency rate is defined.

Assumptions for direct emission calculations :
- Life time of the supermarket is 30 years
- A complete maintenance operation, with refrigerant recovery is performed after 10 years (end of life of display cases)
- Annual servicing, accidental ruptures, and fugitive emissions are presented under a single rate.
- emission rate and recovery efficiency are presented with a lower and an upper threshold

Table 6.1 presents assumptions for direct emission calculation of a typical supermarket with 4400m^2 sales area.

Table 6. 1 assumption for direct emission calculation

Refrigeration system	DX	Distributed	Sec. Loop MPG+ Sec. Loop CO2-	Sec. Loop CO2 + & -	Cascade CO2- Sec. Loop MPG+
R-404A charge (kg)	1600	600	400	400	210
CO2 charge (kg)	0	0	700	1400	450
fugitive emission rates					
upper threshold	30%	25%	20%	20%	20%
lower threshold	18%	15%	12%	12%	12%
Emission rate at initial charge	5%				
recovery efficiency at end of life	lower threshold: 0%		upper threshold: 50%		

The TEWI analysis is calculated for a period of 10 years, corresponding to the refrigeration system life time before refurbishing. GWP of R-404A is 3900 (2006 IPCC assessment report), CO_2 is the reference with a GWP of 1.

6.1 Assumptions for indirect emissions calculation

The energy consumption is calculated for each refrigeration system. The methodology is presented in sections 4 (energy consumption) and 5 (lcca analysis). Table 6.2 summarizes the annual energy consumption for a typical supermarket in CA (LA climatic zone).

Table 6. 2 annual energy consumption per supermarket, for different refrigeration systems

Refrigeration system	DX	Distributed	Sec. Loop MPG+ Sec. Loop CO2-	Sec. Loop CO2 + & -	Cascade CO2- Sec. Loop MPG+
Total energy consumption (kWh/year)	1,570,900	1,493,140	1,743,699	1,648,660	1,553,620

In California, the CO_2 content of one kWh, is dependent upon the energy mix in power generation. Energy Power Mix in California for year 2006 is presented in table 6.3 based on the values of California Energy Commission [CEC07].

Table 6. 3 Energy Power Mix in California in 2006

Energy type	Mix
Coal	28.60%
Large hydroelectric	30.50%
Natural gas	35.40%
Nuclear	0.40%
Eligible renewable	5.10%

Power generation leads to different CO_2 factors, depending on the energy conversion process and the primary energy source. Table 6.4 gives the range of CO_2 content of one kWh produced, for different energy sources [GFE07].

*Carbon equivalent factor is converted in CO2 equivalent factor by the ration of molar masses (MM_{co_2}/MM_c)

Table 6. 4 Emission conversion Factors

Primary energy	Carbon equivalent g/kWh	CO_2 equivalent g/kWh
Gas	100 to 130	367 to 477
Coal	200 to 280	733 to 1026
Hydroelectric	1	3.7
Nuclear	2	7.3
Wind power	2 to 10	7.3 to 36.7

Table 6.5 gives low and high thresholds of the CO_2 factor, taking into account the power energy mix in California. Thresholds correspond to variation in power generators efficiency independently of primary energy source.

Table 6. 5 Calculation of the energy power mix in California

		CO_2 emission factor (g CO_2/kWh)	
Mix (year 2006)		Low threshold	High threshold
Coal	28.60%	209.6	293.4
Large hydroelectric	30.50%	1.1	1.1
Natural gas	35.40%	129.9	168.9
Nuclear	0.40%	0.0	0.0
Eligible renewable	5.10%	0.4	1.9
Averaged CO2 emission factor for year 2004		341.1	465.3

6.2   TEWI calculation

Results of the TEWI calculation for five refrigeration systems are given in table 6.6

Table 6. 6 TEWI calculation

Refrigeration system	DX	Distributed	Sec. Loop MPG+ Sec. Loop CO2-	Sec. Loop CO2 + & -	Cascade CO2- Sec. Loop MPG+
Refrigerant total emissions (metric tonnes)					
R-404A (lower threshold)	3.5	1.2	0.7	0.7	0.3
R-404A (upper threshold)	6.0	2.0	1.1	1.1	0.6
CO2 (lower threshold)	0.0	0.0	1.2	2.3	0.7
CO2 (upper threshold)	0.0	0.0	2.0	4.0	1.3
Direct CO_2 equivalent emissions (Thousands metric tonnes)					
lower threshold	13.7	4.5	2.6	2.6	1.4
upper threshold	23.4	7.7	4.4	4.4	2.3
Total energy consumption (MWh)					
lower threshold	15 709	14 931	17 437	16 487	15 536
Indirect CO_2 equivalent emissions (Thousands metric tonnes)					
lower threshold	5.4	5.1	5.9	5.6	5.3
upper threshold	7.3	6.9	8.1	7.7	7.2
Total CO_2 equivalent emissions (Thousands metric tonnes)					
lower threshold	19.1	9.6	8.5	8.2	6.7
upper threshold	30.7	14.7	12.6	12.1	9.6

DX system has a TEWI of 31,000 metric tonnes CO_2, 2 to 3 times higher than secondary loop systems. GWP of R-404A (3900) is the highest of HFCs currently used, therefore direct emissions contribute to 75 % of DX system TEWI. CO_2 cascade offers the best performances.

Secondary loop and distributed systems may lower the refrigerant charge by a factor of two to four. The lower refrigerant charge can directly decrease emissions in case of ruptures and at equipment end-of-life if a systematic and efficient refrigerant recovery policy is not applied. Indirect and distributed systems lead to significantly shorter refrigerant lines, and thereby limit the number of fittings and brazing. As a consequence, the leak tightness of the system is improved.

Figure 6.1 (lower threshold) and figure 6.2 (upper threshold) illustrate the comparison between direct emissions and indirect emissions over a 10 year lifetime for different refrigeration systems found in typical Californian supermarkets.
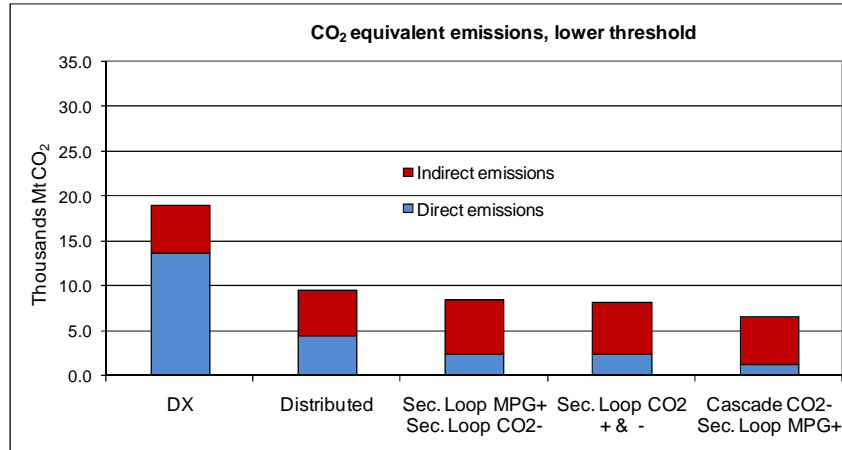
Figure 6. 1 TEWI analysis, lower threshold
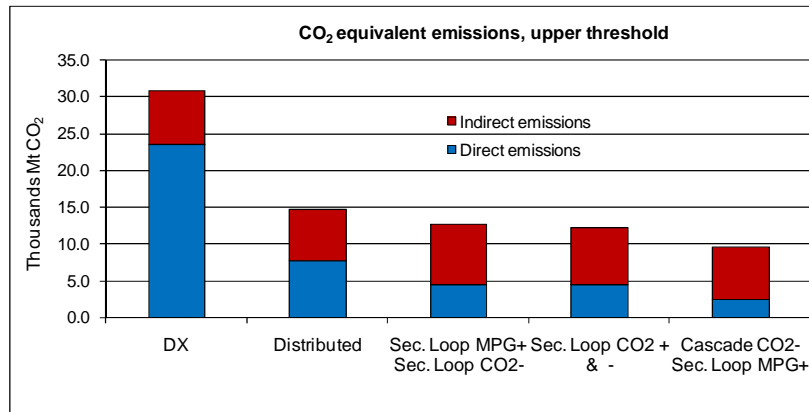
Figure 6. 2 TEWI analysis, upper threshold

6.3 Derivation to Californian state

Taking into account the number of supermarkets in California (~3400), TEWI is calculated at a global level. Table 6.7 and figure 6.3 give the equivalent CO_2 emissions and savings of alternative refrigeration systems, compared with the DX base line.

Table 6. 7 TEWI analysis in Supermarkets at Californian State level

California state	*DX*	*Distributed*	*Sec. Loop MPG+ Sec. Loop CO2-*	*Sec. Loop CO2 + & -*	*Cascade CO2- Sec. Loop MPG+*
Total CO₂ equivalent emissions (Mega metric tonnes)					
lower threshold	64.9	32.6	29.0	27.9	22.6
upper threshold	104.4	49.9	42.7	41.2	32.5
CO2 emissions savings (Mega metric tonnes)					
lower threshold	0.0	-32.3	-35.9	-37.0	-42.3
upper threshold	0.0	-54.5	-61.7	-63.2	-71.9
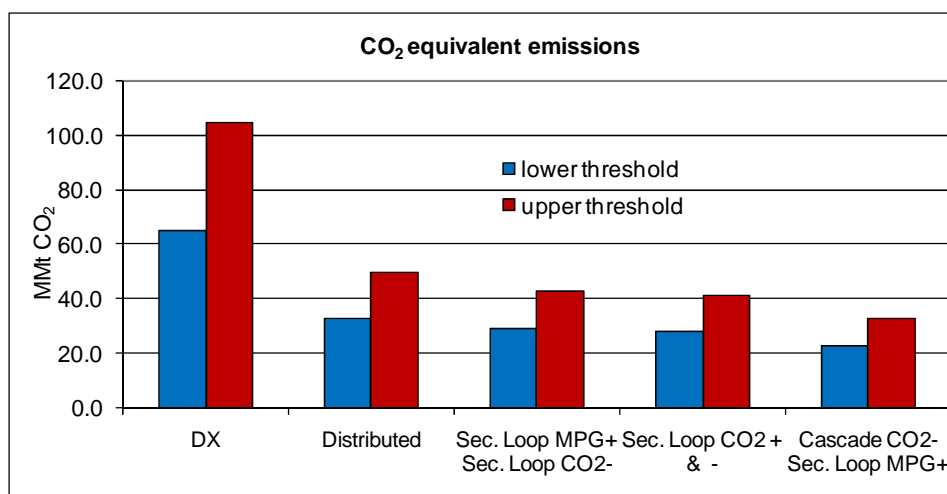
Figure 6. 3 TEWI analysis in Supermarkets at Californian State level

6.5 Costs of CO$_2$ savings

The additional cost for different refrigeration systems has been calculated in the LCCA analysis (cf. chapter 5). Cost of CO$_2$ savings for different refrigeration systems are shown in table 6.8.

Table 6. 8 Costs of CO$_2$ savings

Refrigeration system	DX	Distributed	Sec. Loop MPG+ Sec. Loop CO2-	Sec. Loop CO2 + & -	Cascade CO2- Sec. Loop MPG+
LCCA per supermarket ($)	3,129,341	3,168,385	3,471,194	3,350,382	3,210,353
Additionnal cost ($)	0	39,044	341,853	221,040	81,012
CO$_2$ emission savings (tonnes)	0	-16,040	-18,148	-18,588	-21,145
Cost of 1 tonne CO2 saved ($/metric tonne)		2.4	18.8	11.9	3.8

Combining technical options on display cases and refrigeration system using CO$_2$ cascade and a secondary loop, both direct and indirect CO$_2$ equivalent emissions are reduced. Technical options applied on displays cases are the one described in chapter 4: door installation, led lighting, DC motor and floating head pressure.
Reduction of the cooling capacity has an impact on both energy consumption and refrigerant charge (so refrigerant emissions).

Secondary loop system coupled with CO$_2$ cascade have a major impact on refrigerant direct emissions. CO$_2$ cascade system is more energy efficient than the base line DX system. Savings for technical options applied to display cases, coupled with Cascade CO$_2$ / secondary loop systems are illustrated in table 6.9.

Table 6. 9 Savings for technical options applied to display cases, coupled with Cascade CO_2 / secondary loop systems

Refrigeration system	Per Supermarket		California State	
	DX	Cascade CO2-Sec. Loop MPG+ DC technical options	DX	Cascade CO2-Sec. Loop MPG+ DC technical options
LCCA (Thousands $)	3,129	2,679	10,638,600	9,108,600
CO$_2$ emissions (Thousands metric tonnes)				
lower threshold	19	5	64 894	15 831
upper threshold	31	7	104 412	22 763
CO$_2$ emission savings (Thousands metric tonnes)				
lower threshold		14		49 063
upper threshold		24		81 649

The life cycle cost is 15% lower for a supermarket where technical options for energy savings are applied on both display cases and refrigeration systems. The cut in CO_2 equivalent emissions is 75 – 80% compared to the base line DX system.

7. Refrigerant inventory and emissions

7.1 Method of calculation

The calculation method is described in Annex 2.

7.2 Database for commercial refrigeration in California

The bank of refrigerants contained in grocery supermarkets is calculated based on the last 30 years. The bank represents the cumulated market of refrigerants filled in new refrigerating systems year after year, taking into account the average lifetime of the store (30 years for a supermarket) and the lifetime of the refrigeration system (15 years for centralized systems) before remodeling. The average life time of refrigeration systems is thus 15 years. However, calculations are performed taking into account an extinction curve of the equipment around this average value.

Calculations are performed from year 1990 to 2004, and forecasts are simulated until 2020. In order to initiate calculations as of 1990, the database has to include data from 1960 to 2020, assuming an average 30-year lifetime of grocery supermarkets and other small stores.

7.2.1 Grocery supermarkets

Number of stores in California

The number of grocery supermarkets in California from 1960 to 2020 must be evaluated. Statistical data issued from the US Bureau of Census are used. For a given year, different numbers have been found depending on the source. The definition of supermarket store reported in the US Bureau of Census is as follows:

"Supermarkets and other general-line grocery stores: Establishments commonly known as supermarkets, food stores, grocery stores, and food warehouses, primarily engaged in the retail sale of a wide variety of grocery store merchandise. Customers normally make large, volume purchases from these stores."

Numbers of grocery supermarkets in State of California are available for a few years. More data are available for USA, and can be helpful to evaluate California numbers, by means of ratio based on GDP and population. Those ratios are used to fill the database back to 1960.
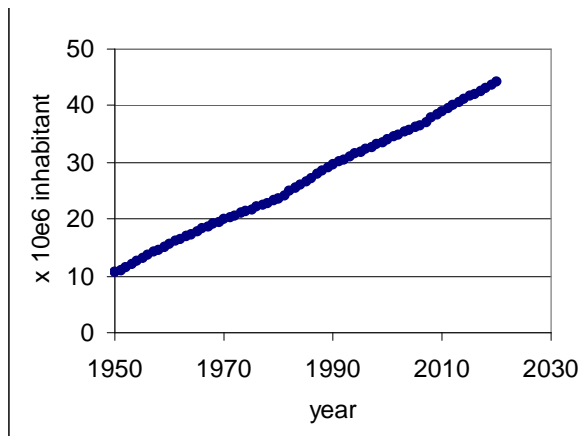
Figure 7. 1 Population growth in California
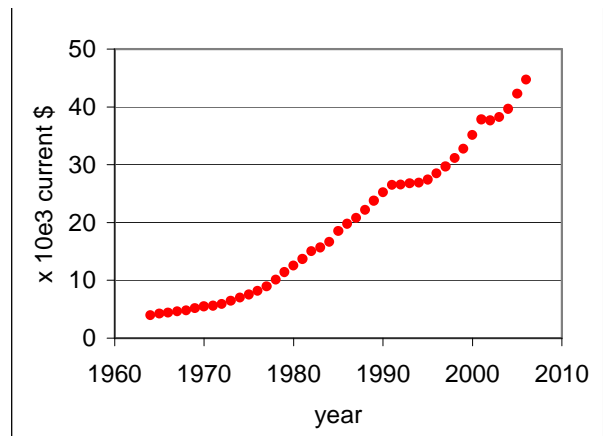
Figure 7. 2 GDP/inhab. growth in California.

Figure 7.3 draws the evolution of supermarket number in California, based on US bureau of census data for US and GDP ratio (California / USA). First supermarkets were built after the world second war. 1950 is taken as the initial year. The average area of supermarket is known for a few years [FMI2007]. A ratio of the total sales area to population was introduced and extrapolated for all the years concerned by this study (see figure 7.4)
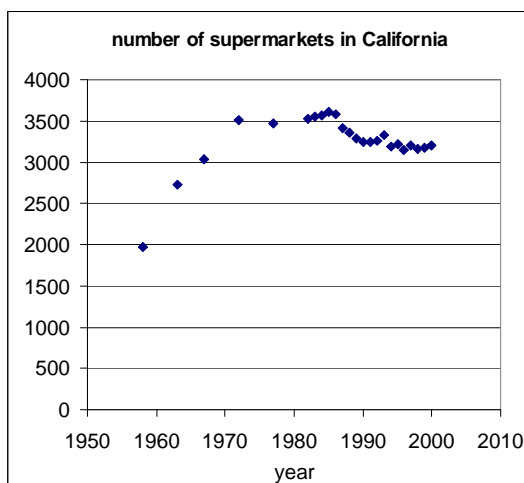
Figure 7. 3 number of supermarket in California. Data from us bureau of census.
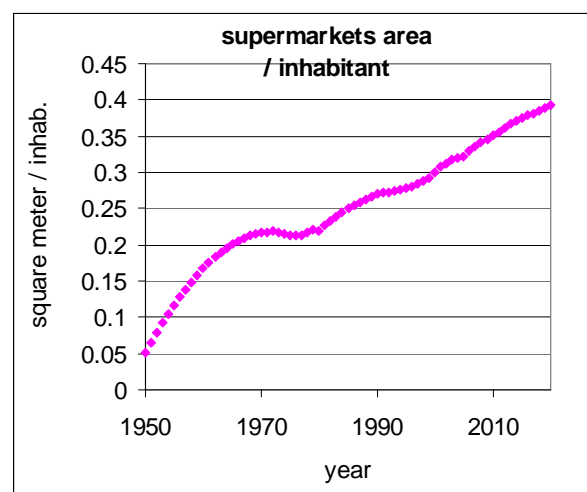
Figure 7. 4 average sale area/per inhabitant growth from 1950 to 2020

Multiplying this ratio (sale area / inhab.) by the population, the total sales area in California is derived from 1950 to 2020 as shown in figure 7.5.

From the evolution of the supermarket number in California, openings of new stores per year can be determined. The average lifetime for a supermarket store is supposed to be 30 years before renewing.
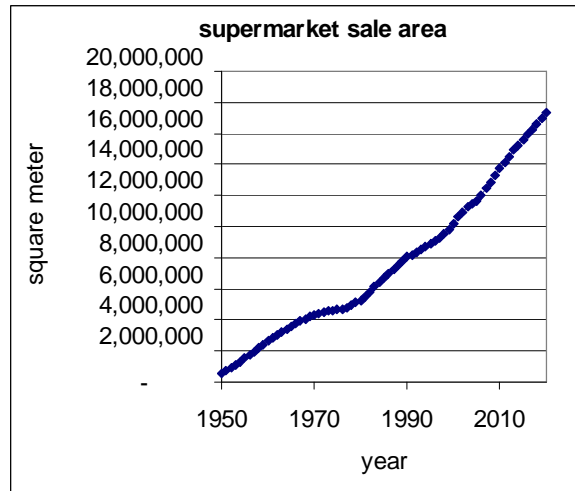
Figure 7. 5 Supermarket total sale area in California.

Average area and refrigerant charge

The average area in 2004 is 4,400 m^2 (47,360 ft^2) but this value was not constant throughout the last 50 years.
Newly opened stores, including supermarkets, have a sales area higher than this average value.

Figure 7.6 draws the evolution of the sales area of grocery supermarkets as of 1950. This value is the typical area of stores classified by vintage, meaning that the average value for all supermarket is lower than the current vintage area.

For example it is assumed that the average sale area of supermarkets in California, whatever their date of opening, is 4400 m^2 in 2004. But a new supermarket opened in 2004 has an average area of 5000 m^2.
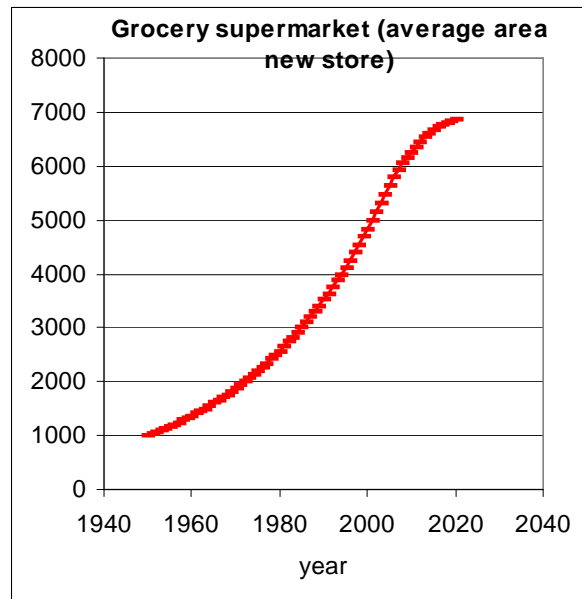
Figure 7. 6 Average area of grocery supermarket.

The refrigerant charge has been determined from the calculation of the cooling capacity of the centralized refrigeration system. The typical layout of the grocery supermarket has been established after the field survey. Each and every display case has been characterized by its cooling capacity and temperature level. Ratios of refrigerant charge per kW of cooling capacity are known for the technology of centralized system. Table 7.1 summarizes the results of calculations used for energy consumption calculations. Table 7.2 gives the ratio of refrigerant charge / cooling capacity for the different technologies of refrigeration systems met in a grocery supermarket.

Table 7. 1 Cooling capacity of each refrigeration system for a typical grocery supermarket.

Cooling Capacity	Medium temperature	Low temperature	Total
Centralized System (kW)	193	152	345 kW
Condensing Units (kW)	17.5	11.7	29.2
Stand-alone (kW)	13.0	1.4	14.5
Total	**263.8**	**138.8**	**402.7**

Table 7. 2 Ratios for refrigerant charge.

Refrigerant charge / Cooling Capacity	Medium temperature	Low temperature
Centralized System in direct expansion	2.8 kg/kW	5.5 kg/kW
Centralized System with secondary loop	0.8 kg/kW	1.2 kg/kW
Condensing Units	1.4 kg/kW	2.4 kg/kW

Stand-alone equipment are not included in this table because the information for the refrigerant charge is known (from the manufacturer) for each type of display case considered.

Using the cooling capacity and the refrigerant charge ratios, the total refrigerant charge for a supermarket is determined. Table 7.3 gives the results. Currently in California, it is considered that the use of secondary loop systems or CO_2 cascade is not significant enough to be considered. On the contrary, forecasts will be performed in order to evaluate the impact of a widespread use of secondary loop systems on refrigerant emissions.

Table 7. 3 Refrigerant charge in a grocery supermarket.

Refrigerant charge	Medium temperature	Low temperature	Total
Centralized System	540 kg	837 kg	1377 kg

During the field survey, in most of supermarket stores visited, it was impossible to visit the machinery room. Nevertheless, some machinery rooms of a few supermarkets have been visited with the store manager. A first cross checking is possible with refrigerant charge indicated in one of these supermarkets. For confidentiality reasons, the brand name and the location of the reference supermarket are not mentioned. Two compressor racks for medium temperature display cases and storage room are charged with 900 lb of R-507A each. The compressor racks for low temperature equipment are charged with 900 lb. This supermarket is representative of the typical grocery supermarkets described (see Section 7.1) and the refrigerant charge is 3 x 450 kg, nearly the same (see Table 7.3) as the one defined using the refrigerant ratio (see Table 7.2) and the cooling capacity (see Table 7.1). Those ratios have been elaborated on a number of field surveys made in the US and in Europe.

For calculations in RIEP, the average refrigerant charge is related to the supermarket sale area and a coefficient of refrigerant charge per square meter is defined. This coefficient, 0.36 kg/m^2 for year 2004, is the expression of the share of refrigerated (and frozen) food in a grocery supermarket. The change in food consumption habits during the last 50 years had an impact on the refrigeration ratio for a given sales area.

The assumption is made that the ratio of refrigerated sales area has increased by 50 % between 1960 and 1990 (see Figure 7.7).

In parallel, the average area of grocery supermarket was increasing (see Figure 7.6), meaning that the refrigerant charge in a typical supermarket was 200 kg in 1960, and is of 1600 kg today.
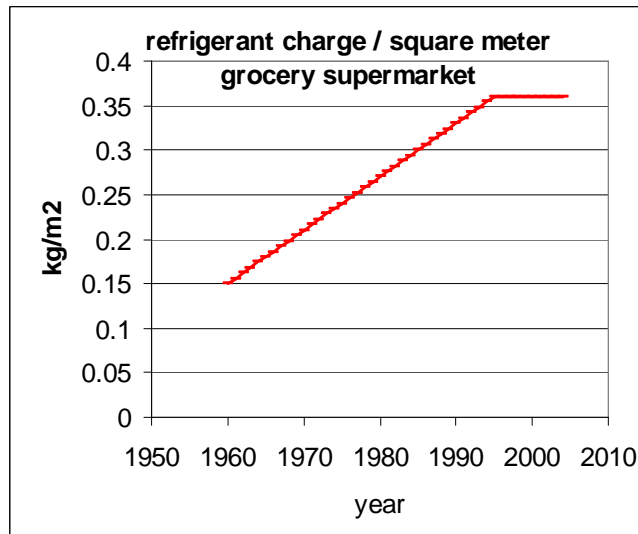
Figure 7. 7 Refrigerant charge ratio (kg/m^2).

Refrigerant emission rates

Refrigerants emissions are of two types: fugitive emissions during all the lifetime of the refrigeration equipment, including accidental emissions (rupture of pressure valve or liquid line), and end-of-life emissions when refrigerant recovery is not performed carefully (if performed).

The fugitive emission rate is an average value taking into account refrigerant losses of different types. This emission rate is established based on refrigerant annual consumption for a given store: the refrigerant quantity refilled annually in the system compensates refrigerant emissions.

Emission rate is estimated at **30 % per year** for centralized systems. This value is a conservative one, emissions could vary from 10 to 30 % [GAG97], [IPC05], [TOC06] but values have to be confirmed along the years. Complementary data are needed but are not easily disclosed by Commercial chains.

Refrigerant recovery at end of life has increased in the past 10 years due to the phase-out of CFCs. Figure 7.8 presents the refrigerant recovery efficiency evolution assumed for RIEP calculations.
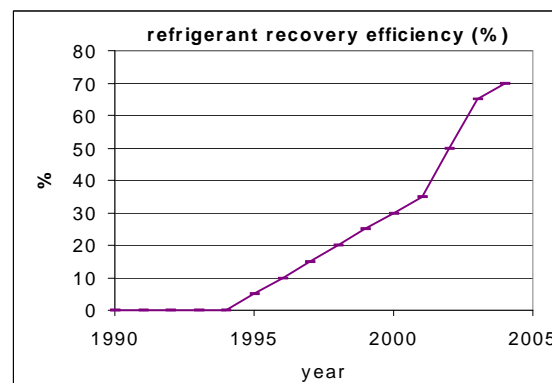
Figure 7. 8 Refrigerant recovery efficiency at end of life.

Refrigerants in use

The refrigerant market share follows the regulations. Since 1995, CFCs are no longer used in new refrigeration systems. CFCs have been replaced first by HCFC blends for retrofitting of existing systems and in new systems. HCFCs will be banned in new refrigeration systems as of

2010. In the US, HFCs start to be used in centralized systems in 1999. Refrigerant shares are presented in Figure 7.9.
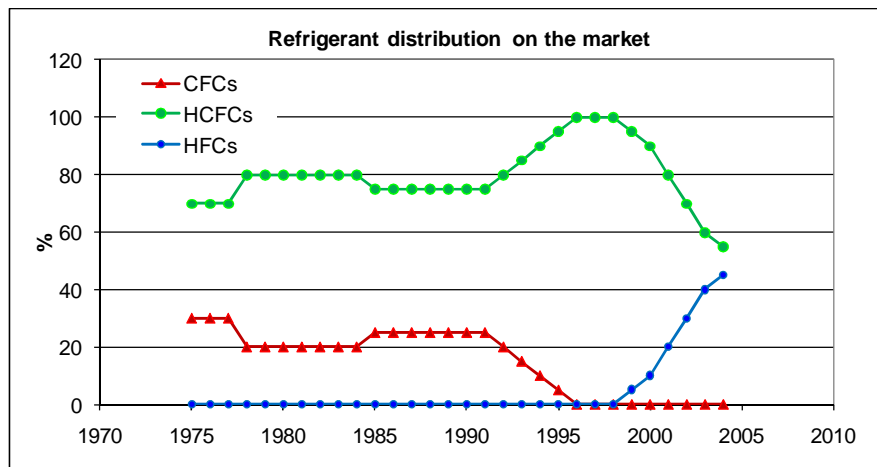
Figure 7. 9 Refrigerant market shares (new equipment and remodeling).

Remodeling of the refrigeration system is supposed to be done at mid lifetime of the store, after 15 years. After 1995, this step is typically the time to retrofit CFCs refrigerant to HCFC blends. Retrofit of CFCs is supposed to be achieved in 2004.

7.2.2 Small stores

Number of stores in California

US bureau of Census data have been mainly used to evaluate the number of the different stores in California since 1960.

The same methodology is applied to each type of store when statistical data are not available. For small stores like bakeries, butcheries, fishmongers, and convenience stores another point has been considered: these small stores were common before the growth of supermarkets. For the last 20 years, the numbers of these small stores has been decreasing. They are replaced by larger sales area stores such as mini-markets, and supermarkets.

Stores, where refrigeration equipment is used, are summarized in Table 7.4. Those stores were described in Section 2 presenting energy consumption in commercial refrigeration.

Table 7. 4 Stores reported in the data base.

Type of stores	Number of stores in 2004
Grocery stores (food dedicated)	3,370
Minimarkets	4,693
Pharmacies	4,846
Convenience stores	2,317
Liquor stores	3,466
Butcheries, Pork-butcheries	763
Fishmonger stores	184
Bakeries and Pastries	5,512
Small size Gas Stations	5,453
Large size Gas Stations	1,818
Hotels	5,458

Motels	5,817
Bars and Restaurants	66,306

Stand-alone equipment studied independently of their location of use	Number of units in use in 2004
Carbonated Soda Fountains	23,040
Vending machines	500,000

Refrigerant charge and type of refrigerant

The field survey has allowed identifying the typical layout of each small store using refrigerating equipment. This refrigerating equipment is not connected to a centralized system in the machinery room like in supermarkets. Stand-alone equipment, display cases and walk-in cooler connected to one or several condensation units are the typical technologies met in those stores. Table 7.5 presents the typical refrigerant charge evaluated for each type of store, based on the field survey.

Table 7. 5 Refrigerant charges.

Type	Refrigerant Charge in stand alone equipments (kg)	Refrigerant Charge in condensing units (kg)
Bakeries	2.65	4.3
Bars & restaurants	2.5	19
Vending machines	0.3	-
Butcheries	0.3	9.7
Center gas stations	3.45	15.8
Convenience stores	4.9	29.2
CSD fountains	0.3	-
Fishmonger stores	0.6	13.4
Grocery supermarkets	7.4	56.5
Hotels	3.4	19.5
Liquor stores	5.8	16.8
Minimarkets	9.3	101.7
Motels	1.45	0
Pharmacies	2.35	37.3
Small gas stations	0.8	3.1

Note: Grocery supermarkets are considered in the list of stores because the use of stand-alone equipment in the sales area is significant. Moreover some walk-in coolers are not connected to the centralized system, but run with independent condensing units.
Stand-alone equipment and condensing units are not similar to centralized system in terms of energy efficiency, but also in terms of refrigerant type and refrigerant emissions. In order to be more accurate in the evaluation of refrigerant emissions, refrigerating equipment has been sorted and calculated by technology: stand alone equipment, condensing unit, and centralized system.

Refrigerant emission rate

Stand-alone equipment is characterized by short refrigerant circuit, the compressor and the condenser being integrated in the cabinet. Tube and fittings are usually brazed permitting to reduce fugitive emissions. Because of a small unitary refrigerant charge (less than 2 kg in most cases), refrigerant recovery at end of life is not done on stand-alone equipment.

Condensing units are more emissive systems. The charge of refrigerant can be significant, for example in mini-markets, where the number of display cases is high. Recovery at end of life should be improved.

Table 7. 6 Emission rates and recovery efficiency.

Type	Emission rate (%)	Recovery efficiency (%)
Stand alone equipment	1	0
Condensing units	15	30

7.3 Refrigerant inventory from 1990 to 2004

7.3.1 Refrigerant demand

The refrigerant demand is the addition of refrigerant needs for servicing of all refrigerating systems in use, and the refrigerant needs for first charge of new refrigerating systems. Figures 7.10 to 7.12 present the refrigerant demand, by type, respectively for centralized systems, condensing units, and stand-alone equipment.
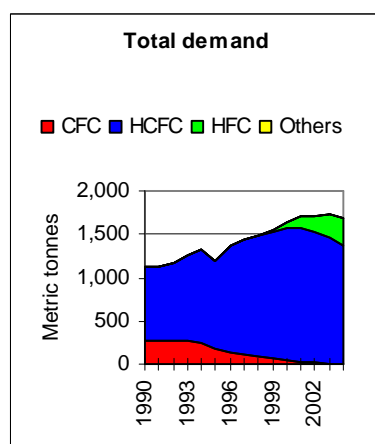
Figure 7. 10 Refrigerant demand of centralized systems (grocery supermarkets).
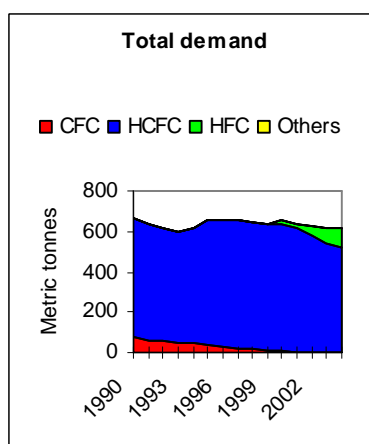
Figure 7. 11 Refrigerant demand of condensing unit systems
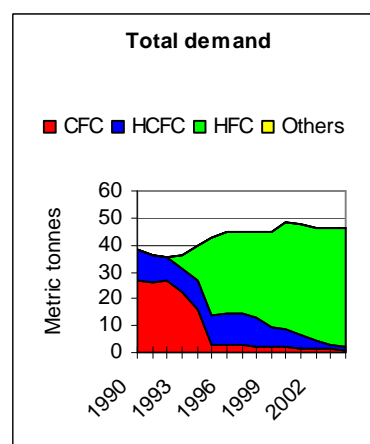
Figure 7. 12 Refrigerant demand of stand-alone equipment.

Depending on the technology (centralized, condensing unit, stand alone) the refrigerant demand is significantly different. As indicated on Figure 7.14, centralized systems with 1,700 tonnes per year, represent nearly 72% of the refrigerant demand in the commercial refrigeration sector.

Refrigerant demand for stand-alone equipment is mainly for new equipment sold on the market, because servicing needs for this technology are low. In terms of refrigerant distribution, HFC are mainly dedicated to stand-alone equipment. In other technologies (centralized and condensing units) refrigerant needs are high for servicing, because of high emission rates of these systems (30% in centralized systems and 15% in condensing units).
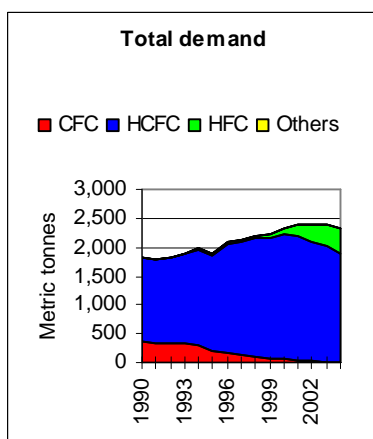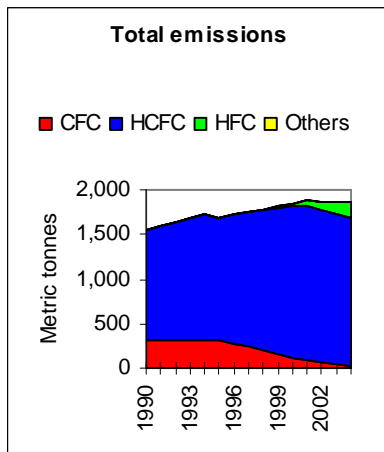
Figure 7. 13 Refrigerant demand in commercial refrigeration.
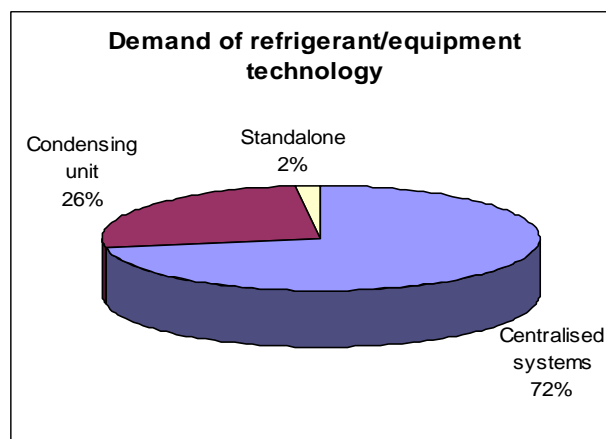
Figure 7. 14 Distribution by refrigeration equipment technology.

HCFC demand, mainly R-22, represents 72% of the refrigerant market. The commercial refrigeration sector demand of HCFCs in 2004 is about 2,000 tonnes.

7.3.2 Refrigerant bank charged in refrigeration equipment

The refrigerant bank is the total amount of refrigerant charged in all refrigeration systems in use whatever their vintage, in commercial sector. Figures 7.15 to 7.17 present the refrigerant bank, by family, respectively in centralized systems, condensing units, and stand-alone equipment.
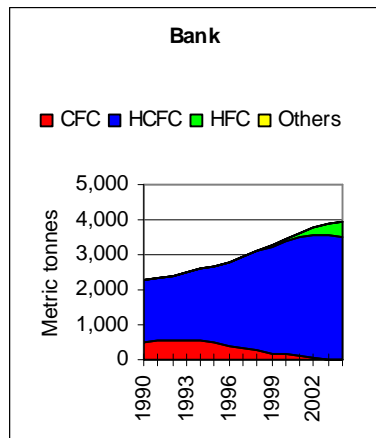
Figure 7. 15 Refrigerant bank in centralized systems (grocery supermarkets).
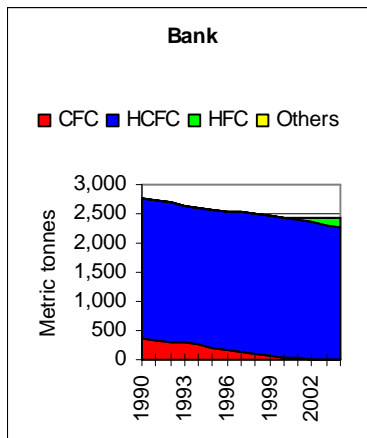
Figure 7. 16 Refrigerant bank in condensing unit systems.
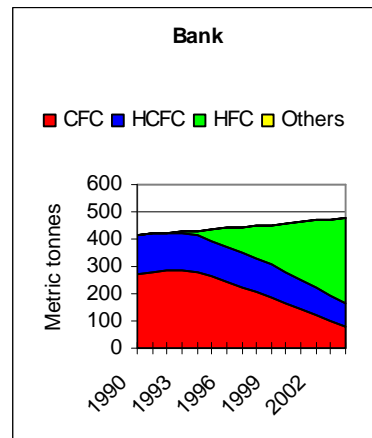
Figure 7. 17 Refrigerant bank in stand-alone equipment.

Bank of refrigerants in centralized systems in supermarkets is nearly 4,000 tonnes in 2004 in California, and 90% of this bank are HCFCs. The introduction of HFC on the market, in new equipment, has begun in 1999. In condensing units, the bank of refrigerant is around 2,500 tonnes, but is not growing any longer.

Stand alone equipment, working with small refrigerant unitary charges, were filled mainly with R-12 before 1992. No retrofit is performed on those systems. In 2004, the remaining bank of R-12 in stand-alone equipment is estimated at 500 tonnes. R-12 has been replaced by R-134a, which is in the main refrigerant in use, today, in stand-alone equipment.

Bank

CFC ■ HCFC ■ HFC □ Others
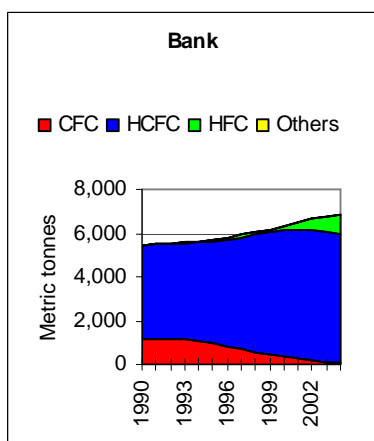
Figure 7. 18 Refrigerant bank in commercial refrigeration.

Bank of refrigerant / equipment technology

Standalone 7%

Condensing unit 35%

Centralised systems 58%
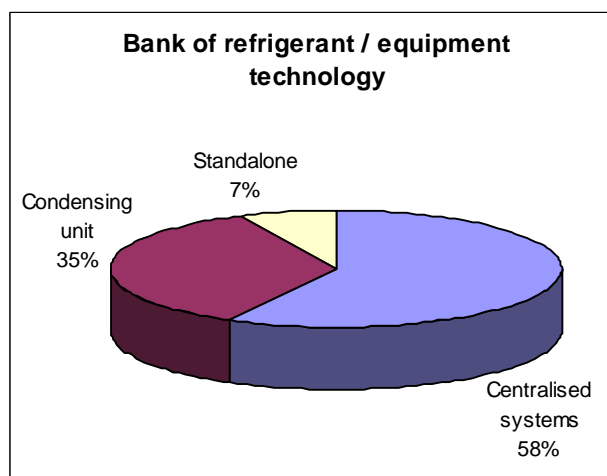
Figure 7. 19 Distribution by refrigeration equipment technology.

Total refrigerant bank in the commercial refrigeration sector is estimated around 6,800 tonnes in 2004, mainly constituted of HCFCs. 58% of refrigerant bank is filled in centralized systems in supermarkets.

7.3.3 Refrigerant emissions

Emissions represent both fugitives losses, and end of life emissions. Figures 7.20 to 5.22 present the refrigerant emissions, by type, respectively from centralized systems, condensing units, and stand-alone equipment.

Total emissions

CFC ■ HCFC ■ HFC □ Others
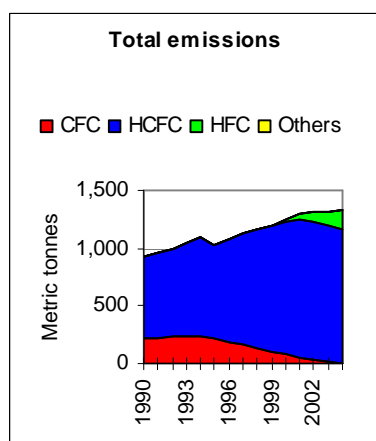
Figure 7. 20 Refrigerant emissions in centralized systems (grocery supermarkets.)

Total emissions

CFC ■ HCFC ■ HFC □ Others
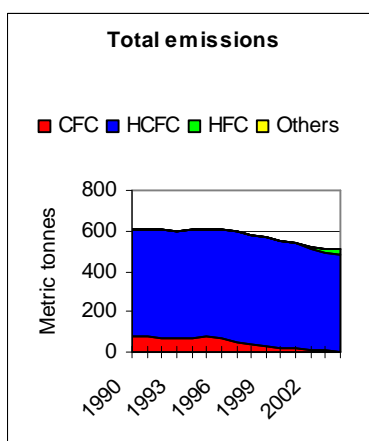
Figure 7. 21 Refrigerant emissions in condensing unit systems.

Total emissions

CFC ■ HCFC ■ HFC □ Others
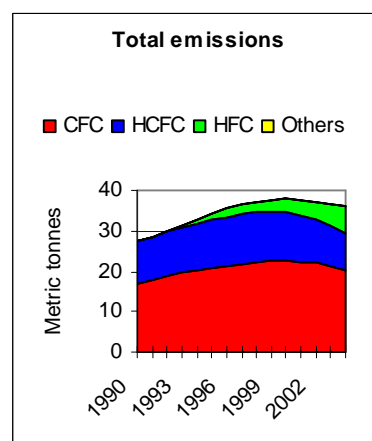
Figure 7. 22 Refrigerant emissions in stand-alone equipment.

Lifetime of these refrigeration systems is estimated to be 15 years in average. For centralized systems, the real lifetime of the system is usually longer. But from1990 to 2020, many retrofit operations are and will be done, because first of the phase-out of CFCs, then followed by the phase out of HCFCs. When the system is renewed, most of the time, refrigerant handling leads to emissions.

Total emissions

CFC ■ HCFC ■ HFC □ Others

Figure 7. 23 Refrigerant emissions in commercial refrigeration.

Refrigerant emissions / equipment technology
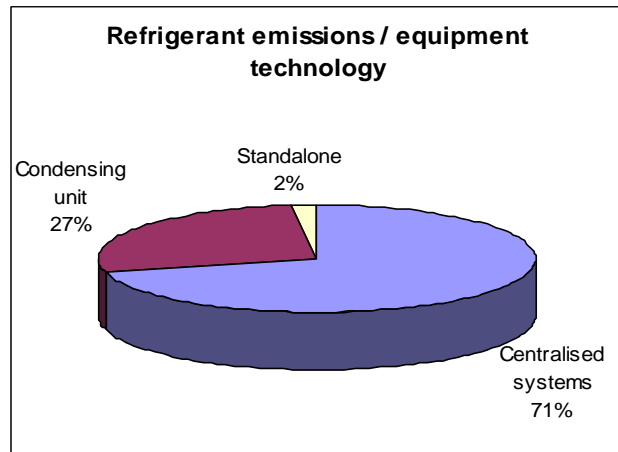
Figure 7. 24 Distribution by refrigeration equipment technology.

Total emissions in commercial refrigeration sector in California are around 1,800 metric tonnes in 2004, coming for 71% from centralized systems.

7.3.4 CO_2 equivalent emissions of refrigerants

Refrigerant emissions expressed in CO_2 equivalent are based on GWP values from the 2nd Assessment Report. Figures 7.25 to 5.27 present the refrigerant emissions in CO_2 equivalent values, respectively from centralized systems, condensing units, and stand-alone equipment.

CO2 Equiv emissions

CFC ■ HCFC ■ HFC □ Others
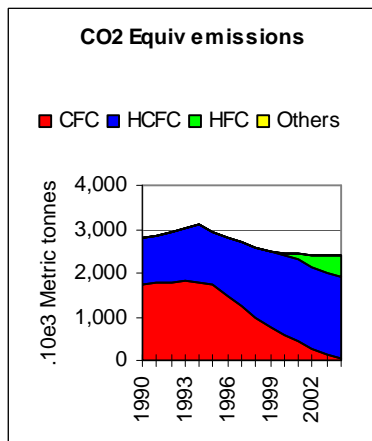
Figure 7. 25 CO_2 emissions in centralized systems (grocery supermarkets).

CO2 Equiv emissions

CFC ■ HCFC ■ HFC □ Others
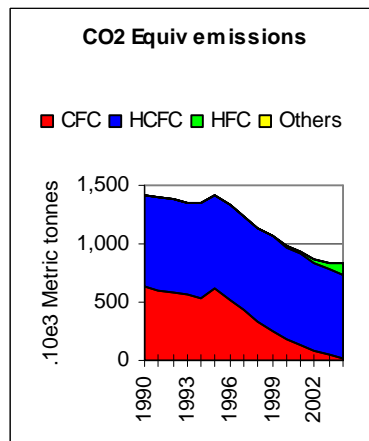
Figure 7. 26 CO_2 emissions in condensing unit systems.

CO2 Equiv emissions

CFC ■ HCFC ■ HFC □ Others
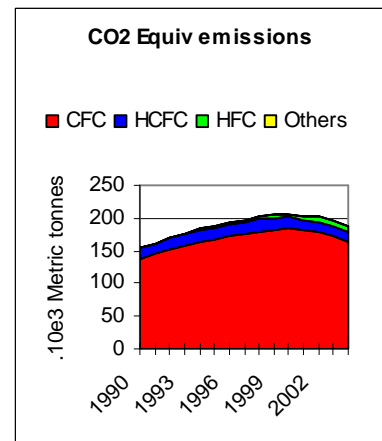
Figure 7. 27 CO_2 emissions in stand-alone equipment.

From 1990 to 1995, CFC emissions represent around 20% of refrigerant emissions in centralized systems in supermarkets. Because of its high GWP (GWP R-12: 8600), emissions of this CFC represent more than 55% of CO_2 equivalent emissions.
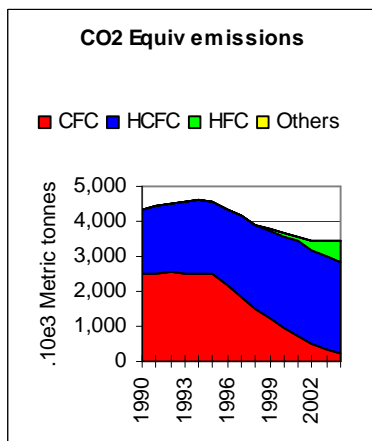
Figure 7. 28 CO$_2$ emissions in commercial refrigeration.
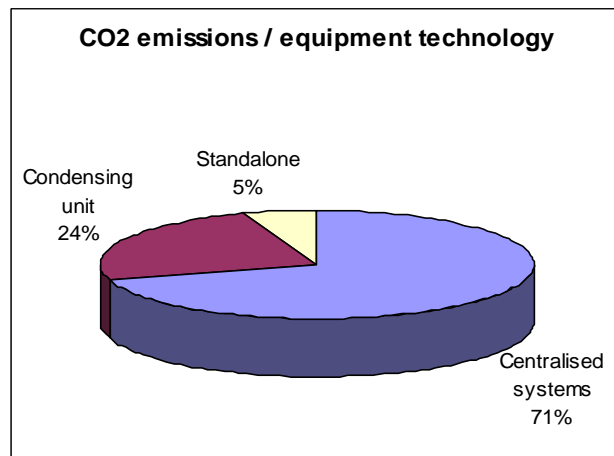
Figure 7. 29 Distribution by refrigeration equipment technology.

Total CO$_2$ equivalent emissions have been decreasing since 1995, because of R-12 phase out. In 2004, CO$_2$ equivalent emissions in commercial refrigeration are more than 3 million metric tonnes CO$_2$. Centralized systems in supermarkets represent 71% of those emissions.

7.4 Scenario and projections to 2020

7.4.1 Assumptions for scenarios

Three scenarios have been simulated to evaluate the impact of technical changes and policies on refrigerants.

Scenario 1: business as usual (BAU)

No significant changes are considered. The regulation organizing the phase-out of HCFCs after 2010 is taken into account. The use of secondary loop systems is not accelerated. Emission rate and recovery efficiency are kept at the same level. There is no significant effort to retrofit R-22 systems.

Scenario 2: large introduction of secondary loop systems

Indirect refrigeration systems decrease the refrigerant charge and minimizing potential refrigerant leakage. Indirect systems have many forms: complete indirect system, partial indirect system, and indirect cascade system. Water solutions have long been used as HTF. Other very promising developments are phase-change HTF mainly CO$_2$.

Starting in 2008, secondary loop systems are progressively introduced. The use of CO$_2$ as HTF is technically possible for both medium and low temperature systems. For now, this technology is only used for low temperature systems, where the pressure is limited to 1.2 MPa in operating conditions.

The assumptions made for the simulations in scenario 2 are as follows:
- 75% of new refrigeration systems are built with a CO$_2$ secondary loop for low temperature applications
- 50% of new refrigeration systems dedicated to medium temperature are secondary loop systems with water solutions of glycols

- Secondary loop systems have an emission rate of 10%, thanks to improved refrigerant containment in the machinery room
- The refrigerant charge is reduced (see Table 7.2) with secondary loop systems, R-404A is the refrigerant used in the machinery room
- R-22 retrofit with R-422A or equivalent intermediate HFC blends starts in 2008 and is totally done in 12 years
- Recovery efficiency is progressively increased to 80%
- The emission rate on new centralized systems is progressively reduced from 30% to 20% thanks to improved leak tightness of components, improvement in the leak search and data reporting when refrigerant losses are observed.

Scenario 3: introduction of low GWP refrigerants, reduction in cooling capacity and refrigerant charge

Recent research on refrigerant has led to new molecule developments, permitting to reach very low GWP refrigerant. In 5 to 10 years, refrigerant blends with GWP lower than 500, for low temperature application, will possibly be available.

Simulations on energy consumption, when all display cases are closed, have shown significant decrease of the refrigeration needs. This scenario is evaluated in scenario 3.

The assumptions taken for the simulations in scenario 3 are:
- Identical to scenario 2, except the choice of R-404A in secondary loop system. A new refrigerant blend, called BLD1(blend 1) with a GWP of 500, is introduced progressively on the market, beginning in 2012. It replaces R-404A and R-507A in new refrigeration systems.
- The cooling capacity is cut by nearly 40%: all open display cases are replaced by display cases equipped with glass doors. The replacement of old display cases is done in 15 years, starting in 2008.

7.4.2 Refrigerant bank filled in refrigeration equipment

Case of centralized system in supermarket only

Figures 7.31 to 5.32 present, for each scenario, the refrigerant bank changes from 2000 to 2020 in centralized systems in supermarkets.
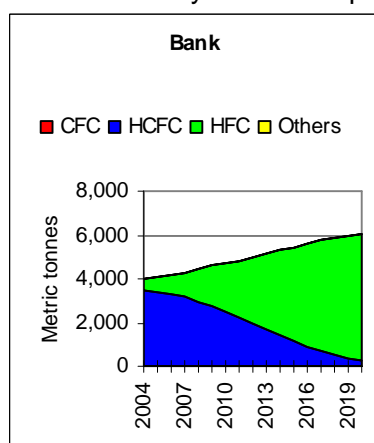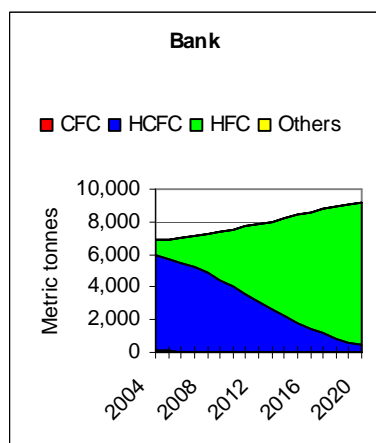

Figure 7. 30 Scenario 1
Refrigerant bank changes.
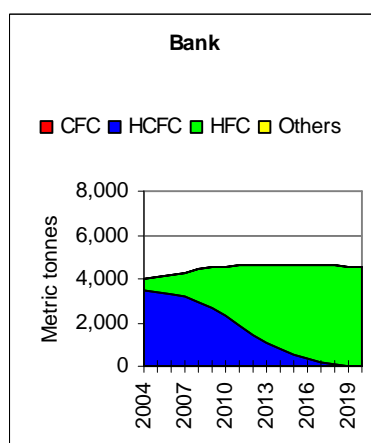
Figure 7. 31 Scenario 2
Refrigerant bank changes.
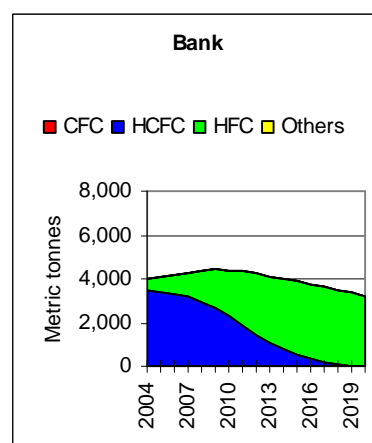
Figure 7. 32 Scenario 3
Refrigerant bank changes.

In scenario 1, business as usual, the refrigerant bank continues to growth, and reaches 6,000 tonnes in 2020. In 2010, HCFC bank is still more than 50% of the total bank.

In scenario 2, the introduction of secondary loop systems starting in 2008, allows reversing the growth of the bank. In 2020, around 2,000 tonnes of refrigerants are avoided compared to the BAU scenario.

In scenario 3, both secondary loop systems, and reduction of the refrigeration needs in the sales area (glass doors) have permitted to divide the BAU bank of refrigerants by a factor 2, at 3,000 tonnes, in 2020.

Commercial refrigeration sector, including small stores

Figures 7.33 to 5.34 present, for each scenario, the refrigerant bank changes from 2000 to 2020 in the commercial refrigeration sector.

In the business as usual scenario, the total bank of commercial refrigeration sector reaches 9,000 tonnes in 2020.

In scenario 2, the impact of secondary loop introduction in supermarkets is less significant in relative value, because of the refrigerant bank in condensing units. Technically, the use of secondary loop systems is possible in small stores in replacement of condensing units, but the uptake of the technology is relatively slow.

In the commercial refrigeration sector, including small stores, the impact of measures taken in scenarios 2 and 3 are significant. In scenario (3), the refrigerant bank reduction is 30% in 2020.

Figure 7. 33 Scenario 1
Refrigerant bank changes.
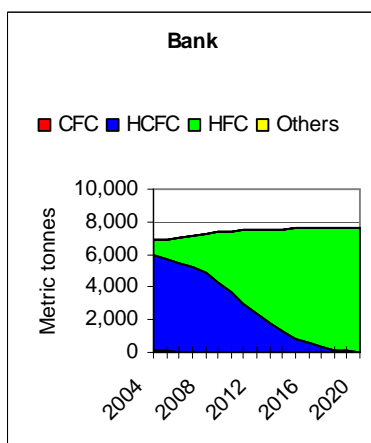
Figure 7. 34 Scenario 2
Refrigerant bank changes.
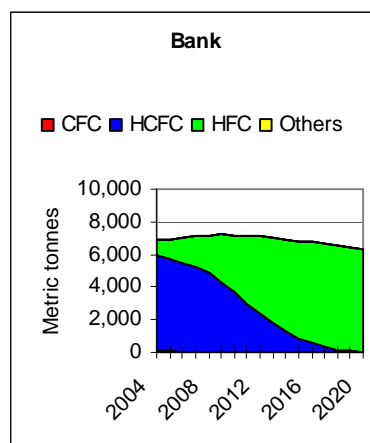
Figure 7. 35 Scenario 3
Refrigerant bank changes.

7.4.3 Refrigerant emissions

Centralized systems in supermarkets only

The impact of changes in technology, moving to secondary loop systems, is both on the refrigerant charge and on the fugitive emission rate. As shown on Figure 7.37 (scenario 2) and Figure 7.38 (scenario 3), the level of HFC emissions is divided by 2 (scen. 2) and divided by a factor 3 in scenario 3.
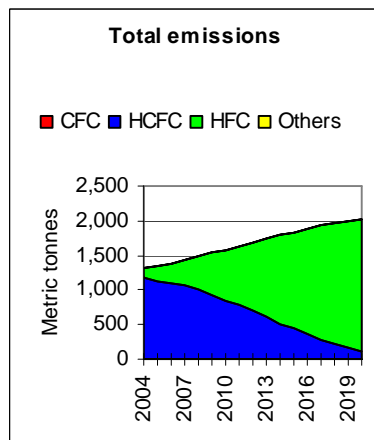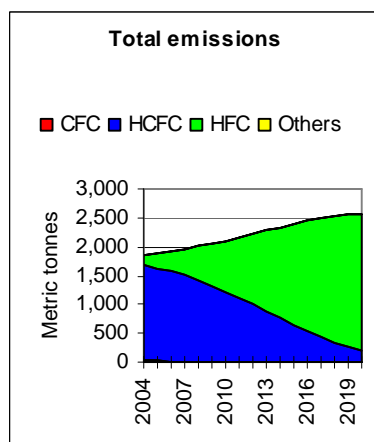
Figure 7. 36 Scenario 1
Refrigerant emission changes.
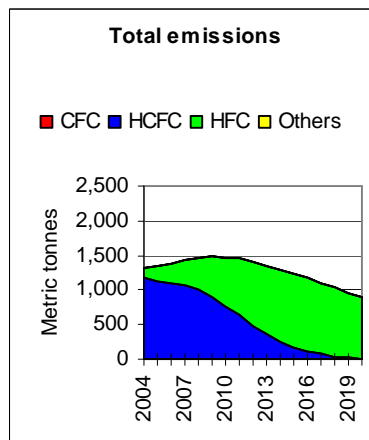
Figure 7. 37 Scenario 2
Refrigerant emission changes.
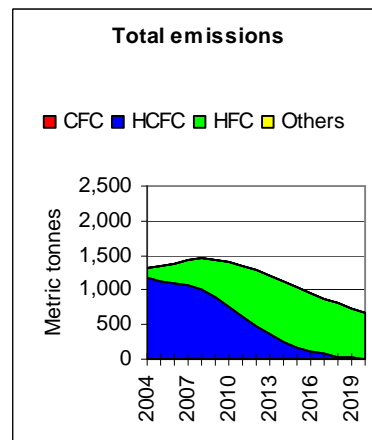
Figure 7. 38 Scenario 3
Refrigerant emission changes.

Commercial refrigeration sector, including small stores

In scenarios 2 and 3, improvement of leak tightness and of recovery efficiency have also been considered for condensing units and stand-alone equipment. Figures 7.39 to 5.41 present the results in emission reductions for the commercial refrigeration sector, taking into account all technologies.

Figure 7. 39 Scenario 1
Refrigerant emission changes.
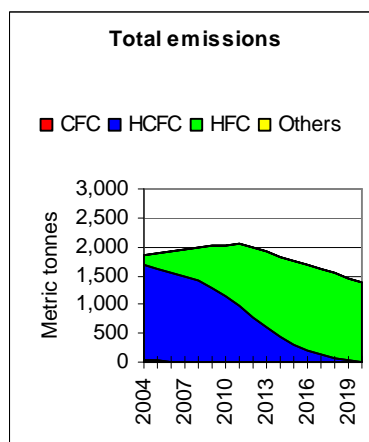
Figure 7. 40 Scenario 2
Refrigerant emission changes.
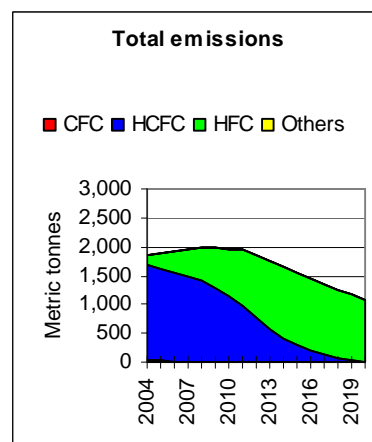
Figure 7. 41 Scenario 3
Refrigerant emission changes

In the business as usual scenario, the level of refrigerant emissions is above 2,500 tonnes per year in 2020. In scenario 2, after the introduction of secondary loop systems in supermarkets, refrigerant emissions are limited to 1,400 tonnes in 2020. When the cooling capacity is decreased (scenario 3), in addition of a secondary loop system, refrigerant emissions are lower: 1,000 tonnes in 2020.

7.4.4 Refrigerant CO_2 equivalent emissions
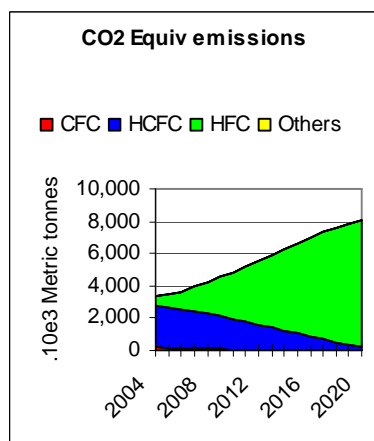
Centralized system in supermarkets only
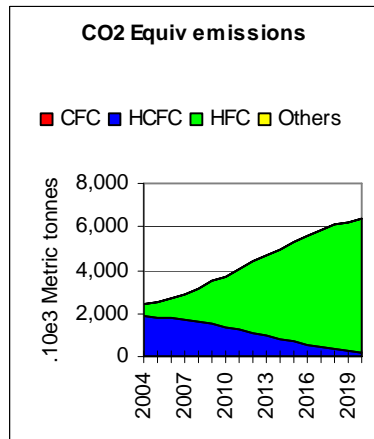
Figure 7. 42 Scenario 1
CO_2 emission changes.
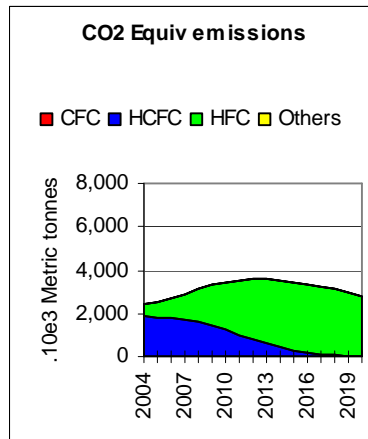
Figure 7. 43 Scenario 2
CO_2 emission changes.
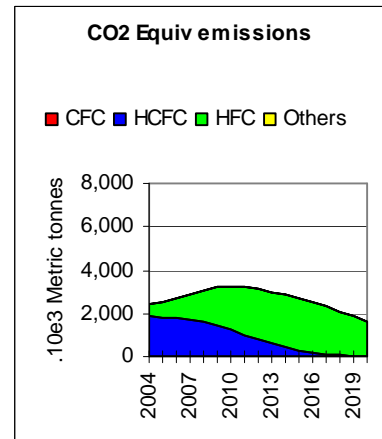
Figure 7. 44 Scenario 3
CO_2 emission changes.

R-507A and R-404A have the highest GWP of HFC refrigerants currently used. The phase-out of HCFCs that had lower GWPs has an impact on CO_2 equivalent emissions. In scenario 1, which is the current scenario for year 2000 to 2008, a minimum of CO_2 equivalent emissions is observed in 2004. After this date, the wide use of R-404A has a negative impact on CO_2 equivalent emissions. Those emissions could reach 6.2 millions tonnes in 2020, more than the values met in the period of use of CFCs (from 1990 to 1994) (see Figure 7.25).

Commercial refrigeration sector, including small stores

Figure 7.45. Scenario 1
CO_2 emission changes.
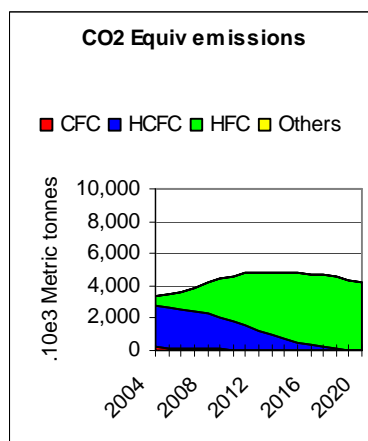
Figure 7.46. Scenario 2
CO_2 emission changes.
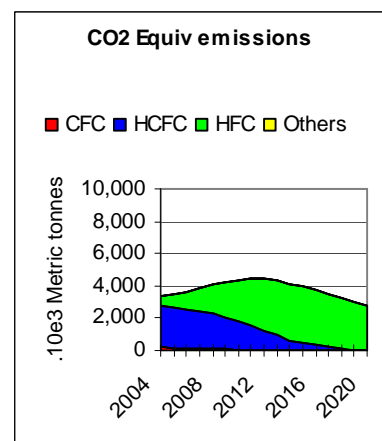
Figure 7.47. Scenario 3
CO_2 emission changes.

Changes in refrigerants, with low GWP blends, are supposed to start in 2012, only on new refrigeration systems. The simulation of scenario 3 does not consider a retrofit of existing systems with R-404A. Nevertheless, the reduction in CO_2 equivalent emissions is significant: less than 2.5 million tonnes in 2020, instead of 8 millions in scenario 1.

7.4.5 HCFC recovery and refrigerant demand

In 2010, the production of virgin HCFCs is banned. In developed countries, the use of HCFCs is still possible with recycled fluids, but the demand will have an impact on the refrigerant prices. Figures 7.48 and 7.49 give an evaluation of HCFC demand for servicing in commercial refrigeration, and, in parallel, the total amount of HCFCs recovered after end of life or retrofit operation.
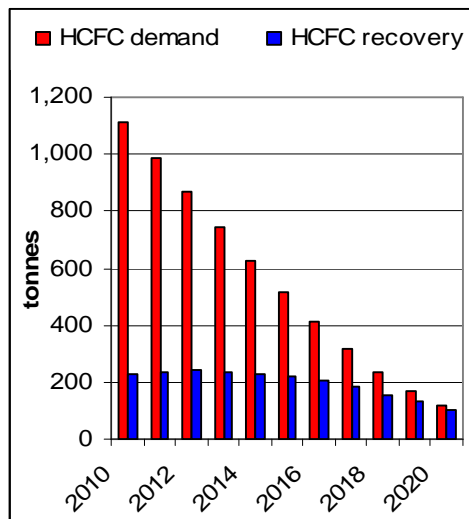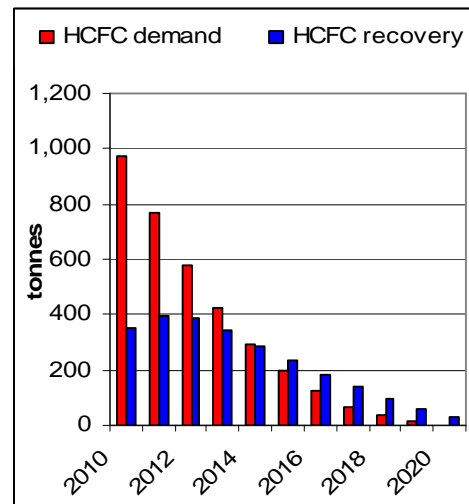

Figure 7.48. Scenario 1
R-22 demand and recovery.

Figure 7.49. Scenario 2 & 3
R-22 demand and recovery.

In scenario 1, no retrofit has been considered before the end of life of systems, or before the renewing period (15 years). Figure 7.48 shows clearly the lack in refrigerant for the period from 2010 to 2020: the recovery of R-22 is nearly 400 tonnes per year, when the refrigerant needs for servicing are 1800 tonnes in 2010 and 700 tonnes in 2015. Without any leak tightness improvement, and retrofit policy, HCFC needs will exceed available recycled refrigerants and thereby the change to intermediate blend has to be highly accelerated.

In scenarios 2 and 3 (Figure 7.49), retrofits of HCFC installations start in 2008 and generate additional refrigerant on the market after recycling. The demand for HCFCs is covered by recovery from 2013 to 2020. Before 2013, needs of R-22 for servicing can not be covered by refrigerant recovery from the commercial sector only.

8. Unitary Air Conditioning and chillers

This section cover two sub-domains:
- air to air stationary air-conditioning systems
- chillers

Chillers are used for climate comfort and in industrial processes. Chiller manufacturers consider that about 2/3 of large chillers manufactured have been installed for climate comfort. The two equipment sub-domains and even the eight categories of air/air AC systems and the two categories of chillers exist as specific parts in the RIEP database. These categories have been merged into one category when it comes to markets, banks, emissions, etc… in order to comply with the IPCC reporting format and limit the amount of tables and figures and.

8.1 Air to air systems

8.1.1 Data sources and detailed calculation method

Data for sales and production of new equipment are not available for California. Therefore, a ratio (*Population of California/Population of USA*) is applied to available statistics of the USA from BSRIA [BSR02], [BSR05]

$$Market\ California\ (year\ i) = Market\ USA\ (year\ i) * Ratio\ (year\ i) \qquad (8.1)$$

$$Production\ California\ (year\ i) = Production\ USA\ (year\ i) * Ratio\ (year\ i) \qquad (8.2)$$

This source includes 8 categories of air-to-air systems:
- Portable/Moveable
- Window
- Splits (Ductless <5kW)
- Splits (Ductless > 5kW)
- Indoor Packaged
- Ducted Splits < 17kW
- Ducted Splits > 17kW
- Roof top.

In the RIEP database, the eight different categories are calculated separately, and one global methodology is applied to all categories. Details concerning differences in refrigerant charge and choices are given the following sections.

Based on data from reference sources, the equipment production and markets are calculated, taking into proper account exports and imports of equipment. Knowing the average charge and the refrigerant type selected for each eight air-to-air AC category, the annual refrigerant quantity charged in new equipment is calculated. This also applies to the total refrigerant charge of the equipment exported.
- With data on the annual refrigerant market and the equipment lifetime, the Californian refrigerant bank can be determined.
- Using the fugitive emission rate of each category, the annual refrigerant servicing market of a country is determined.
- Refrigerant emissions (fugitive and at end of life) can be derived from the refrigerant bank while using data on the lifetime of the equipment.

8.1.2 Installed base of AC Unitary systems

Based on data available for each category [TOC03] [TOC07] [BSR02] [BSR05], the average refrigerant charge can be established (see Table 8.1). These values correspond to average values if one uses information on the typical shares of refrigerating capacities (which is directly related to the refrigerant charge) within the different categories.

Table 8. 1 – Characteristics for the eight categories of air-to-air AC equipment

Type	Charge (kg)	Life (years)
Portable/Moveable	0.5	10
Splits (Ductless <5kW)	1	15
Splits (Ductless > 5kW)	7.5	15
Indoor Packaged	5.5	15
Window	0.7	12
Roof Top	21	20
Ducted Splits < 17kW	3.5	15
Ducted Splits > 17kW	9	15

Table 8.1 shows as well the average lifetime used to establish the law of end of life of equipments given in the following figures:
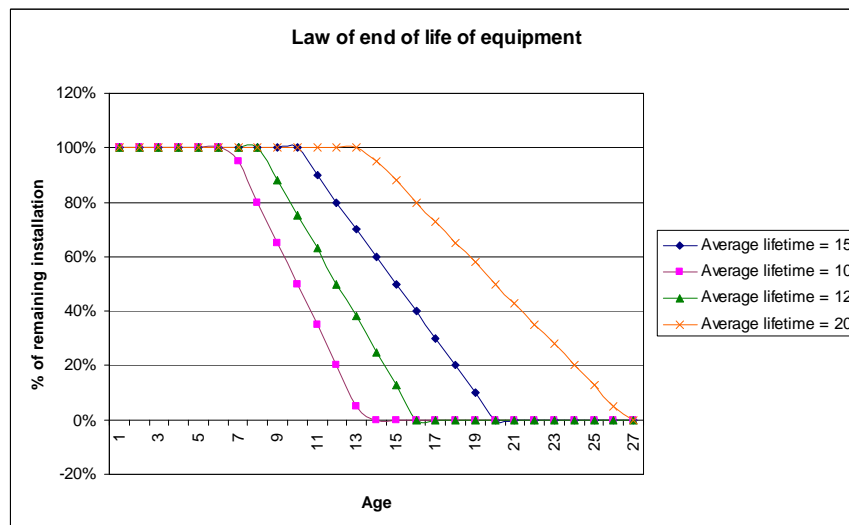
Figure 8. 1: law of end of life equipment (cf. annex 3)

End of life curves given above are applied to the market of corresponding AC unitary system. The installed base for eight categories are given below:

Table 8. 2: Installed base of stationary air conditioners

| Year	Portable	Splits (Ductless <5kW)	Splits (Ductless > 5kW)	Indoor Pack	Window	Roof Top	Ducted Splits < 17kW	Ducted Splits > 17kW
1990	11,820	39,290	18,011	418,596	3,598,786	785,746	3,288,974	332,330
1991	12,680	42,460	19,464	452,398	3,875,957	856,228	3,544,786	359,164
1992	13,580	45,750	20,972	487,456	4,160,071	929,383	3,807,006	386,999
1993	14,460	49,120	22,516	523,378	4,447,229	1,004,331	4,081,997	415,517
1994	15,420	52,660	24,136	561,036	4,748,529	1,082,784	4,366,097	445,414
1995	16,380	56,310	25,809	599,925	5,059,829	1,163,782	4,662,766	476,289
1996	17,340	60,020	27,761	639,127	5,384,157	1,250,271	5,005,851	508,741
1997	18,300	62,810	30,959	678,525	5,734,143	1,336,451	5,342,049	540,454
1998	19,260	65,750	34,311	718,222	6,097,200	1,422,595	5,667,471	572,244
1999	20,240	68,790	37,876	758,109	6,480,214	1,509,996	5,997,946	618,732
2000	21,260	71,930	41,619	798,209	6,856,471	1,598,626	6,335,109	665,973
2001	22,620	75,810	44,204	835,867	7,065,471	1,682,353	6,658,249	706,990
2002	24,340	81,930	45,617	870,958	7,525,829	1,760,888	7,026,360	741,621
2003	26,400	89,620	45,851	903,462	8,031,314	1,834,038	7,368,209	769,781
2004	28,520	97,950	46,095	940,395	8,410,257	1,907,035	7,740,046	800,630
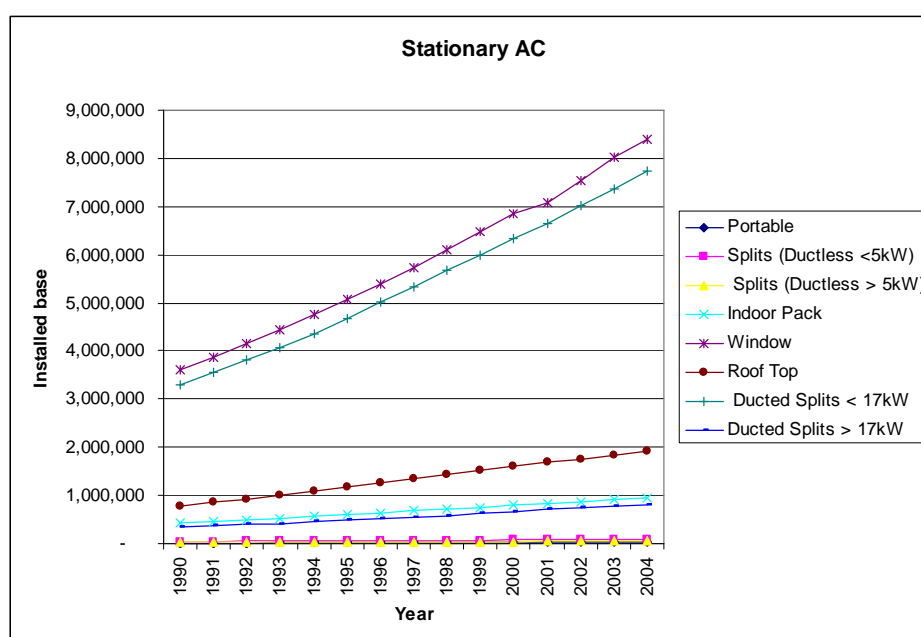
Figure 8. 2: installed base of stationary air conditioner

8.1.3 Analysis of the uptake of R-410A

The phase out of HCFC in brand new equipment in Europe and Japan have lead to the introduction of HFC blends on the market : R-410A and R-407C.
In the US, the use of R-22 in a new equipment is allowed until year 2010. Since little interest is shown for using R-407C, and R-22 is still widely used. Nevertheless, the use of R-410A, a higher pressure refrigerant and a nearly azeotrope blend, makes it possible to reduce the compressor size and increase the compactness of the AC unit.

R-410A units were introduced on the market in 1999 but were limited to small size units. R-410A compressors are specially designed for high operating pressures, up to 40 bar. At first, the range of R-410A compressor capacity was limited to 25 kW cooling capacity. Later on, researches and developments were carried out and allowed in 2004, to reach an average cooling capacity of 100 kW for a roof top with R-410A.

In the US and California, the share of used in brand new equipment approached 10% in 2004. However, a rapid growth is forecast and R-410A will be the major refrigerant used for stationary AC.

8.1.4 Fugitive emission rate and recovery efficiency

Table 8.3 summarizes the fugitive emission rate and the recovery efficiency at end of life for stationary equipments. The recovery efficiency is directly related to the refrigerant charge. Namely, the higher the refrigerant charge is, the higher the recovery rate will be. This may sound optimistic, but corresponds to the thresholds in a number of regulations, i.e., recovery is made mandatory above a certain refrigerant charge value.

Table 8. 3: fugitive emission rates and recovery efficiency in 2004

	Portable	Ductless Splits<5kW	Ductless Splits>5kW	Indoor Packaged	Window	Roof Top	Ducted Splits<17kW	Ducted Splits>17kW
Emission rate	2%	5 %	10%	5%	2%	5%	5%	5%
Recovery efficiency	0%	0%	30%	50%	0%	70%	50%	70%

8.2   Chillers

8.2.1 Data sources and calculation method

An additional parameter to take into account when dealing with chillers is the refrigerant charge ratio, is used here in the calculation process. The refrigerant charge ratio depends on the type of chiller: volumetric (reciprocating, screw and scroll) or centrifugal. These two categories exist as specific separate parts within the RIEP database.

Data for California are not available. Market data are derived from the BSRIA marketing study [BSR05]. A ratio (*Population of California/Population of USA*) is applied to the US available statistics. The average cooling capacity of U.S. chillers is 3 MW and the average cooling capacity of a volumetric chiller is 350 kW. Table 8.4 shows the chiller market in 2004

Table 8. 4: Chiller market in 2004 [BSR05]

Chillers market	USA	California
Centrifugal	3402	418
Other Chillers	14980	1845

8.2.2 Installed base of Chillers

The installed base of chillers is the derivation of chiller market for the last 20 years, taking into account the extinction curve of these units.

Table 8. 5- installed base of chillers, from 1990 to 2004

Year	Centrifugal chillers	Other chillers (volumetric)
1990	9,309	13,553
1991	9,465	13,942
1992	9,781	14,389
1993	10,430	14,941
1994	11,360	15,687
1995	12,268	16,310
1996	13,122	17,240
1997	13,571	18,175
1998	13,571	18,833
1999	13,748	19,247
2000	13,982	19,737
2001	14,145	20,295
2002	14,246	20,886
2003	14,281	21,500
2004	14,309	22,175

8.2.3 Analysis of the uptake of R-123

Refrigerants used in centrifugal chillers (CFC11, CFC-12, HFC-134a and HFC-123) are significantly different from those used in volumetric chillers (HCFC-22, HFC-134a, R-410A, and ammonia). For these two categories, refrigerants types are subsequently analyzed below.

Table 8. 6: Centrifugal chillers, refrigerant distribution in market of brand new equipment

Year	% of R-134a	% of R-11	% of R-12	% of R-123
1990	0	70	30	0
1991	0	70	30	0
1992	5	70	25	0
1993	10	50	20	20
1994	20	30	10	40
1995	40	0	0	60
1996	40	0	0	60
1997	40	0	0	60
1998	40	0	0	60
1999	40	0	0	60
2000	40	0	0	60
2001	40	0	0	60

2002	40	0	0	60
2003	40	0	0	60
2004	50	0	0	50

Table 8. 7: volumetric chillers, refrigerant distribution in market of brand new equipment

Year	% of R-134a	% of R-717	% of R-22	% of R-410A
1990	0	2	98	0
1991	0	2	98	0
1992	0	2	98	0
1993	1	2	97	0
1994	2	2	96	0
1995	3	2	95	0
1996	3	2	95	0
1997	3	2	95	0
1998	3	2	95	0
1999	3	2	95	0
2000	3	2	95	0
2001	3	2	95	0
2002	3	2	95	0
2003	3	2	95	0
2004	3	2	90	5

8.2.4 Refrigerant charge, emission factor and recovery efficiency

The refrigerant charge corresponding to the refrigerating capacity varies significantly with the technology and the liquid density. The ration of refrigerant charge per one kilowatt of refrigerating capacity is shown in Table 8.8 for centrifugal chillers.

Table 8. 8: Charge / cooling capacity ratio for centrifugal chillers [TEA04].

Refrigerant	CFC-11	CFC-12	HCFC-123	HFC-134a
Charge/cooling capacity ratio (kg/kW)	0.28	0.35	0.40	0.36

When considering volumetric chillers, two technical options are available for the evaporator, either dry expansion evaporator (dry Hex) or flooded evaporator used with centrifugal chillers, the. As shown in Table 8.9, the evaporator technology influences the refrigerant charge per kW of refrigerating capacity.

Table 8. 9: Charge / cooling capacity ratio for volumetric chillers [TEA04].

Refrigerant and chiller type	Evaporator type	kg/kW
HCFC-22 and HFC-134a screw and scroll	Dry Hex	0.27
R-410A and R-407C scroll	Dry Hex	0.27
HCFC-22 and HFC-134a screw	Flooded	0.35
HCFC-22 reciprocating	Dry Hex	0.26
R-717 screw or reciprocating	Dry Hex	0.04 to 0.2
R-717 screw or reciprocating	Flooded	0.2 to 0.25

Table 8. 10: emission rate for centrifugal chillers

Centrifugal chillers	R-134a	R-11	R-12	R-22	R-123
Emisson rate (%)	5	10	10	10	3

Table 8. 11: recovery efficiency

Chiller type	Centrifugal chillers	Volumetric chillers
Recovery Efficiency (%)	80	50
Emission rate (%)	(See table 8.10)	5

8.3   Results of calculations: refrigerant bank and emissions

8.3.1 Refrigerant bank

Figures 8.3 and 8.4 present refrigerant bank evolution from 1990 to 2004 in stationary air conditioning and chillers.
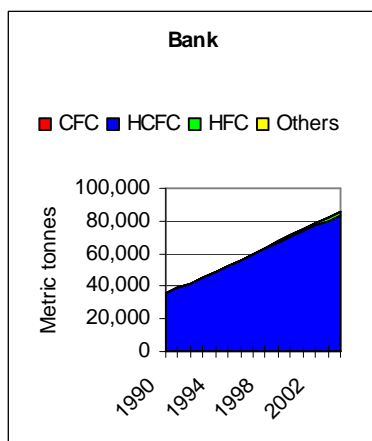
Figure 8. 3: refrigerant bank in stationary AC
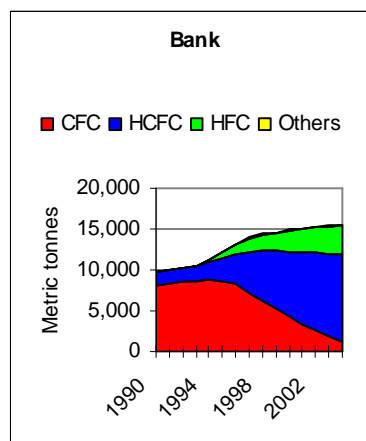
Figure 8. 4: refrigerant bank in chillers

R-22 dominates refrigerant bank in stationary AC sector where a remarkable growth rate is observed between year between 1990 and 2004. In 2004, the installed base of stationary equipment is filled with more than 80,000 tonnes of R-22.

Chillers contribute to 16,000 tonnes of refrigerant bank, which represent 80% of HCFCs in 2004, R-123 in centrifugal chillers and R-22 in other technologies. CFC bank is still positive in 2004 but will be null in a couple of years.

8.3.2 Refrigerant emissions

Figures 8.5 and 8.6 illustrate refrigerant emissions evolutions from 1990 to 2004 in stationary air conditioning and chillers.
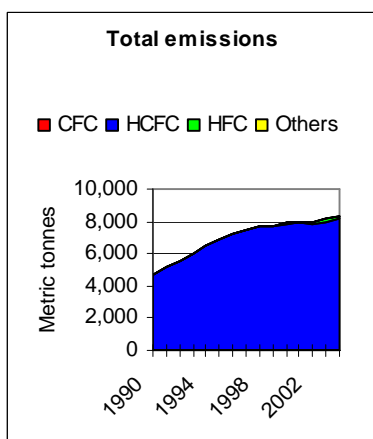
Figure 8. 5: refrigerant emissions in stationary AC
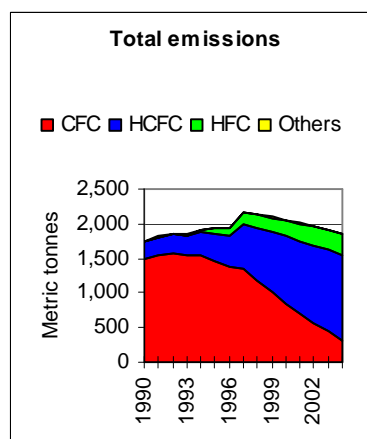
Figure 8. 6 refrigerant emissions from chillers

Following the trend of the bank, the emissions in stationary AC sector reach 8,000 tonnes in 2004. CFC emissions from chillers is significant with nearly still 500 tonnes in 2004.

8.3.3 Refrigerant CO_2 equivalent emissions

Figures 8.7 and 8.8 present refrigerant CO_2 equivalent emissions from 1990 to 2004 in stationary air conditioning and chillers. CO_2 equivalent emissions from chillers were significant in the beginning of the 90s due to the high GWP of R-12 and R-11 compared to R-134a and R-22. In stationary AC sector CO_2 equivalent emissions totalize 12 million of metric tones CO_2.
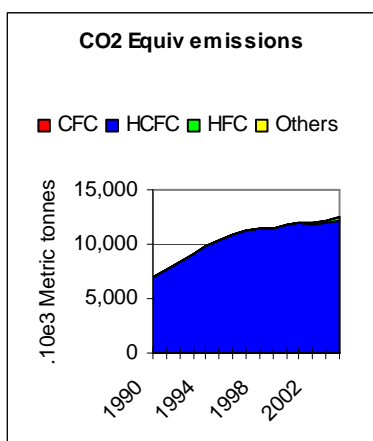
Figure 8. 7: CO_2 equivalent emissions from stationary AC
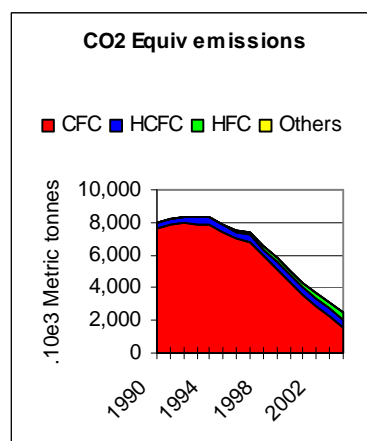
Figure 8. 8: CO_2 equivalent emissions from chillers

8.4 Scenarios to forecast refrigerant emission from unitary air conditioners and chillers

8.4.1 Scenarios assumptions

Business as usual scenario (scenario 1): Scenario 1: Business As Usual

Usual practices and emission rates are kept unchanged for the next 16 years. The recovery efficiency is not improved. Nevertheless, the regulation corresponding to the refrigerant phase out is taken into account for refrigerant replacement.
No effort is made to perform the retrofit of HCFCs during this period.

Scenario 2:

Some improvements are made to reduce equivalent CO_2 emissions of refrigerants:
- The system leak tightness is improved by choosing more reliable components.
- The recovery efficiency is improved at servicing and end of life. Recovery begins on low charge equipment where it was not done before.
- Technologies permitting to reduce the refrigerant charge are chosen (compactness, Dry HX.
- Lower GWP refrigerants are preferred when possible.
- Retrofit of R-22 starts from 2012 for a period of 12 years.

Scenario 3: Partial phase-out of high GWP HFCs

Efforts are made with the same improvements mentioned in the second scenario. Technological options are chosen in order to decrease refrigerant charge and GWP when possible. A new blend is introduced with a low GWP (100). In 2020, this blend will cover 50% of market of brand new equipment.

Table 8. 12: fugitive emission for different scenarios

Emission rate (%)	2004	2020 - Scenario1	2020 - Scenario 2	2020 - Scenario 3
Portable	2	2	2	2
Splits (Ductless <5kW)	5	5	5	5
Splits (Ductless > 5kW)	10	10	8	8
Indoor Packaged	5	5	5	2
Window	2	2	2	2
Roof Top	5	5	5	5
Ducted Splits < 17kW	5	5	5	5
Ducted Splits > 17kW	5	5	5	5

The level of emission rate is low in stationary AC equipment, compared to commercial and industrial refrigeration. Unitary AC systems are compact and most of components are molded. The number of fittings is limited and the sensitivity to leaks is low.

Table 8. 13: recovery efficiency

Recovery efficiency (%)	2004	2020 - Scenario1	2020 - Scenario 2	2020 - Scenario 3
Portable	0	0	30	30
Splits (Ductless <5kW)	0	0	30	30
Splits (Ductless > 5kW)	30	30	50	50
Indoor Packaged	50	70	70	80
Window	0	0	30	30
Roof Top	70	70	70	80
Ducted Splits < 17kW	50	50	70	70
Ducted Splits > 17kW	70	70	70	70

Recovery of refrigerant at the end of life is not performed on low charge equipment as windows, portable or split systems. In scenario 2 and 3, it is considered that refrigerant recovery starts in 2010 for these small unitary systems.

Table 8. 14: refrigerant charge ratio

Ratio kg/kW	2004	2020 - Scenario1	2020 - Scenario 2	2020 - Scenario 3
Centrifugal chillers	0.3	0.3	0.25	0.25
Volumetric chillers	0.35	0.35	0.3	0.3

Flooded evaporators have a large content of refrigerant while dry expansion heat exchangers are more adapted to reduce the refrigerant charge. Moreover, air cooled condenser can be built with micro channel flat tubes to limit the refrigerant charge.

8.4.2 Refrigerant bank

Figures 8.5 and 8.6 present refrigerant bank evolution from 1990 to 2004 in stationary air conditioning and chillers. No significant change in the bank growth is observed for different scenarios. In 2010, new systems are no more filled with HCFCs. R-22 is replaced by HFCs and especially R-410A.
Figure 8.9 shows that the bank growth 2020 in stationary AC units is still high and could reach 150,000 tonnes. In chillers the bank stabilizes at 15,000 tonnes. the growth of the market is absorbed by the decrease of the refrigerant charge in new chillers.
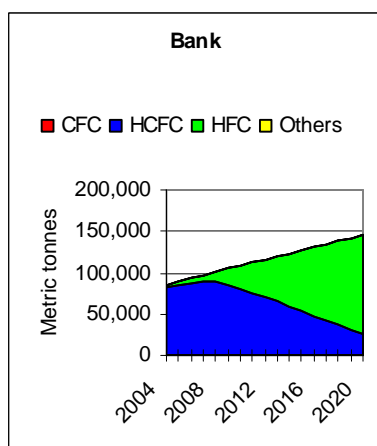
Figure 8. 9: bank in stationary AC
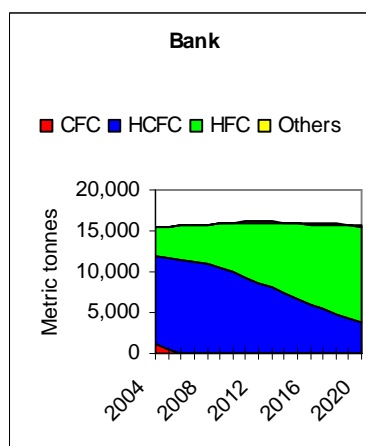
Figure 8. 10: bank in chillers

8.4.3 Refrigerant emissions

Figures 8.11 to 8.13 present refrigerant emissions evolution from 2004 to 2020 in stationary air conditioning.
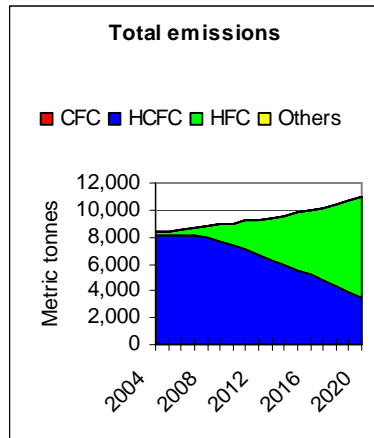
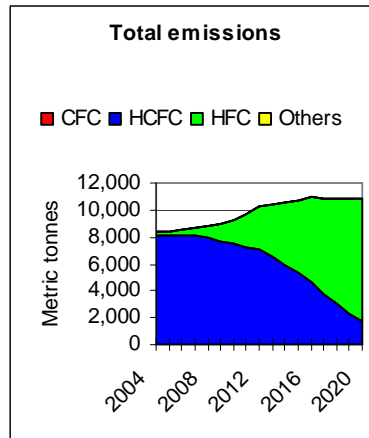Figure 8. 11: scenario 1, emissions from stationary AC

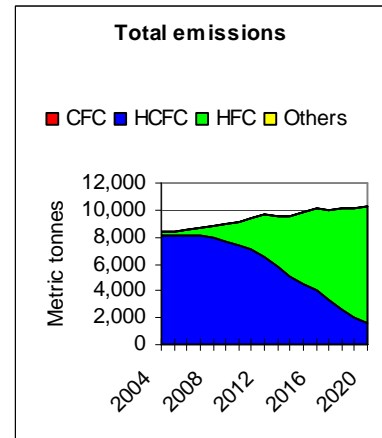Figure 8. 12: scenario 2, emissions from stationary AC

Figure 8. 13: scenario 3, emissions from stationary AC

In 2004, the level of refrigerant emissions (fugitive and end of life) is 8,000 metric tonnes for stationary AC and consists mainly of HCFCs. In BAU scenario, refrigerant emissions reach 11,000 in 2020 due to the bank increase.

In scenario 2 and 3, a better recovery of refrigerant is considered at the equipment end of life. Nevertheless, the side effect of R-22 retrofit causes a rapid increase of HCFC emissions after 2010. Retrofit of R-22 generates anticipated end of life emissions. In 2020, HCFC emissions are lower and the total amount of refrigerant release to the atmosphere is limited at 10,000 metric tones. Figures 8.14 to 8.16 present refrigerant emissions evolution from 2004 to 2020 in chillers.
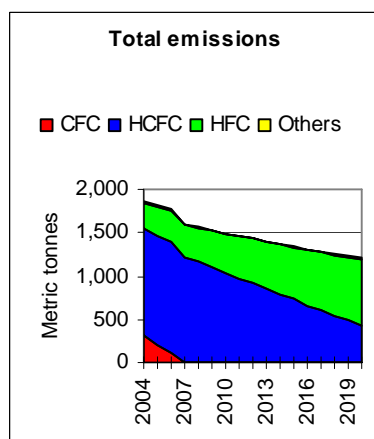
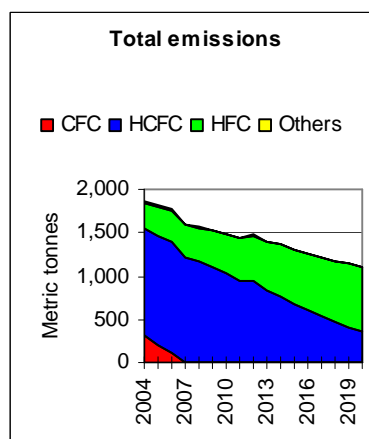Figure 8. 14: scenario 1, emissions from chillers

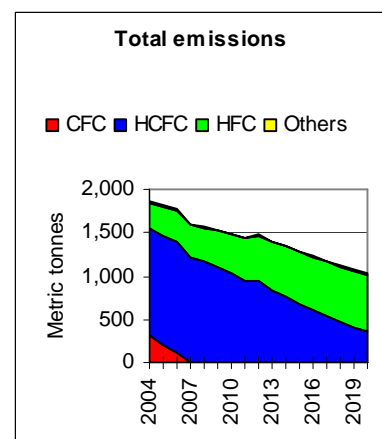Figure 8. 15: scenario 2, emissions from chillers

Figure 8. 16: scenario 3, emissions from chillers

The new generation of chillers, filled with R-134a and R-123 are more leak tight than CFCs chillers. These improvements, along with a better recovery efficiency at the end of life, leads to a reduction in the refrigerant emissions level to 1,000 metric tones in 2020.

8.4.4 Refrigerant CO_2 equivalent emissions

Figures 8.17 to 8.19 present CO_2 equivalent emissions evolution from 2004 to 2020 in stationary air conditioning sector.
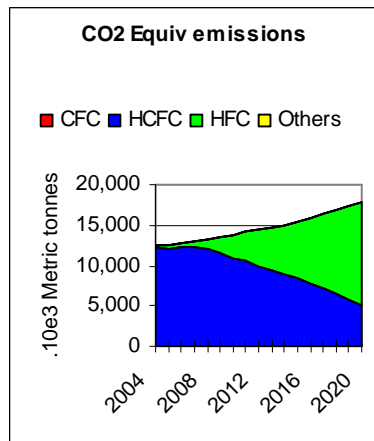

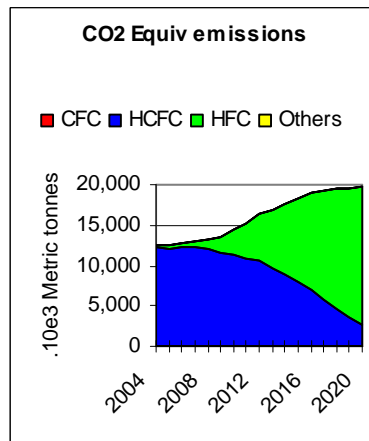Figure 8. 17: scenario 1, CO_2 equivalent emissions in stationary AC

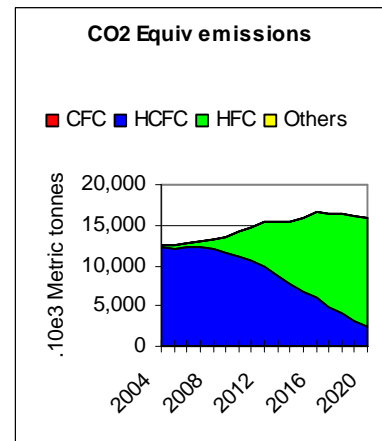Figure 8. 18: scenario 2, CO_2 equivalent emissions in stationary AC

Figure 8. 19: scenario 3, CO_2 equivalent emissions in stationary AC

R-22 GWP is 1500, R-410A GWP is nearly 2000. The introduction of R-410A on the market has an impact on CO_2 equivalent emissions in stationary AC. As shown in scenario 2, the retrofit of R-22 increases CO_2 equivalent emissions, because of higher GWP of HFCs. In 2020 the level of emission reaches 20 million metric tonnes CO_2 equivalent.

In scenario 3, a new refrigerant blend with a very low GWP is assumed available in 2012, and filled in 50% of new equipment in 2020. Therefore, CO_2 equivalent emissions in 2020 will decrease to 15 Million metric tonnes CO_2 equivalent. Figures 8.20 to 8.22 present CO_2 equivalent emissions evolution from 2004 to 2020 in chillers
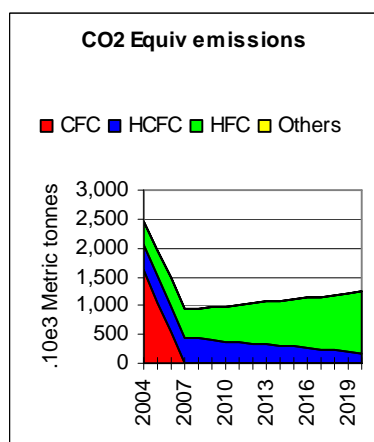

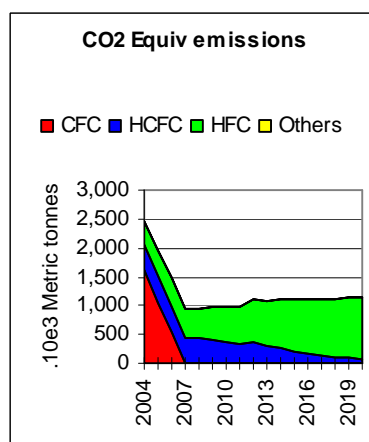Figure 8. 20: scenario 1, CO_2 equivalent emissions in chillers

Figure 8. 21: scenario 2, CO_2 equivalent emissions in chillers
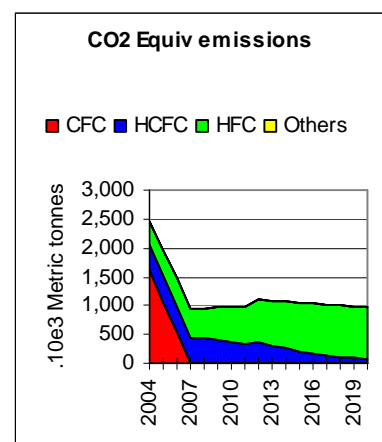

Figure 8. 22: scenario 3, CO_2 equivalent emissions in chillers

The observation concerns the phase out of CFCs in 2007 in centrifugal chillers. This fact has an impact on CO_2 equivalent emissions. R-123 is a low GWP refrigerant, and R-134a GWP is "only" 1300 (compared to R-12 (GWP 8600) and R-11 (GWP 4600)). In scenario 3, the level of CO_2 equivalent emissions is limited to 1 million metric tonnes in 2020.

8.5 Refrigerant demand and recovery

Figures 8.23 and 8.24 present the comparison of R-22 demand recovery for the period 2010-2020 in both stationary AC and chillers sectors.
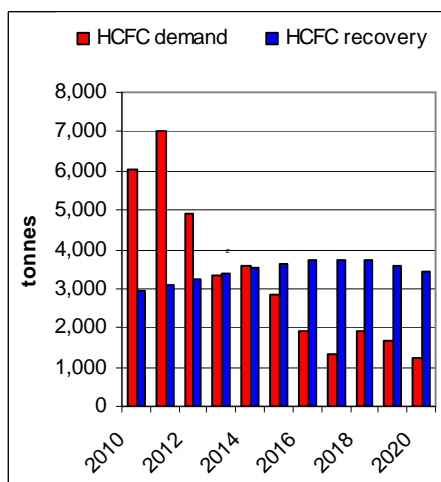

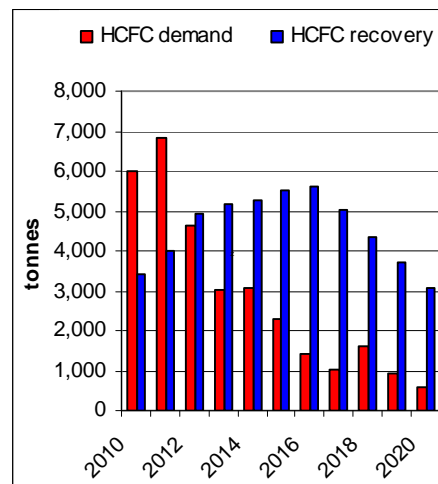Figure 8. 23: HCFC demand and recovery in business as usual scenario

Figure 8. 24: HCFC demand and recovery in scenario 2

In Business As Usual (BAU) scenario (cf. Figure 8.23), HCFC demand from 2010 to 2012 is twice the recovery of HCFCs for this period. After this period, the recovery of HCFC is enough to feed the market for servicing demand .

In scenario 2, retrofit of R-22 in large capacity equipment starts in 2010, for a period of 15 years. In 2010 and 2011, R-22 recovery covers 2/3 of servicing demand, after 2012 the recovery exceeds the demand.

These scenarios show that the phase out of R-22 could not be a problem in stationary and chiller sectors if the recovery is efficiently carried out in these sectors.

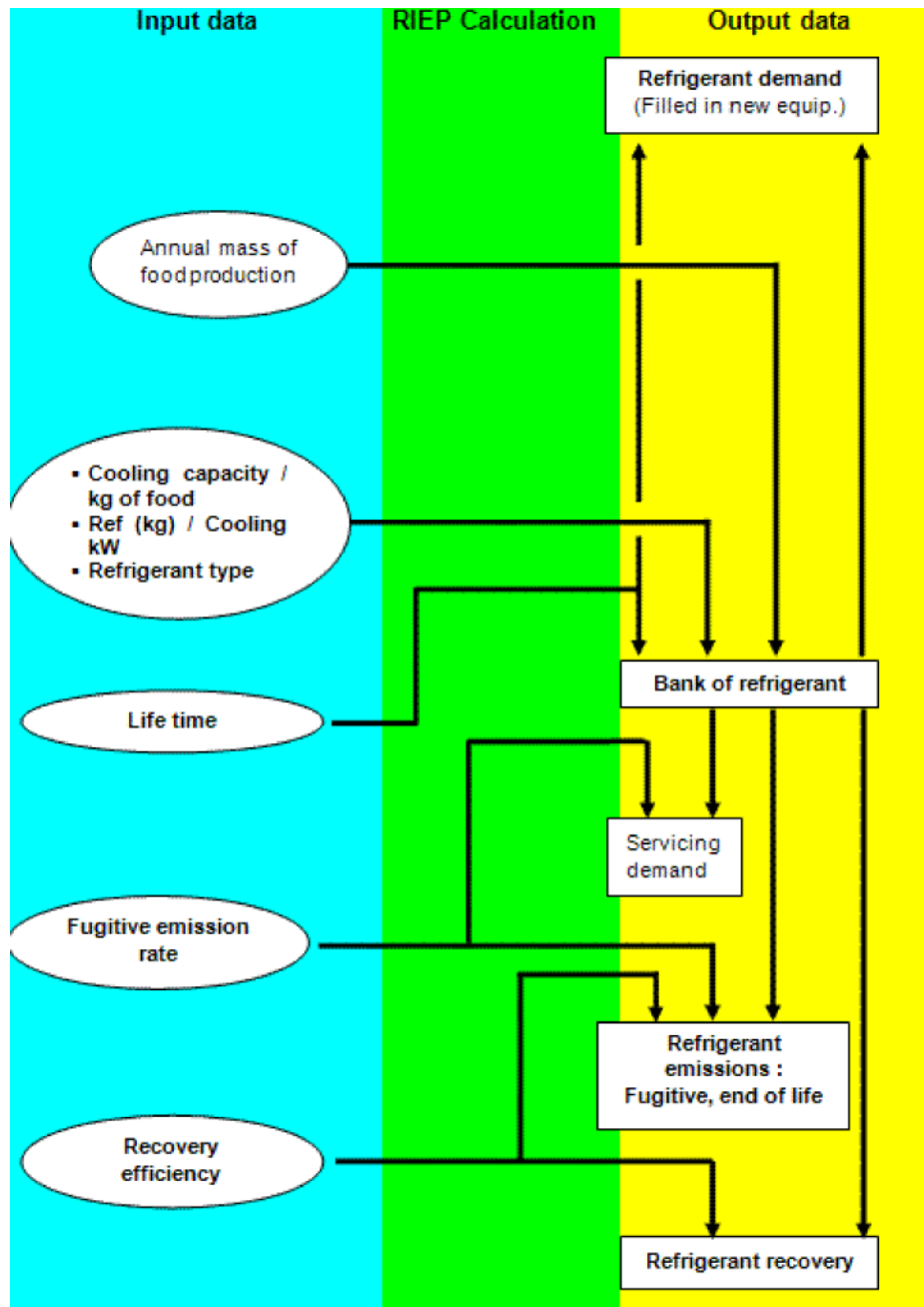# 9. Refrigerant emissions from the industrial sector

Figure 9.1 – Calculation steps for emission forecasts in the food industry.

9.1 Food industry and cold storage

Cooling and freezing processes in the food industry are applied to all types of meat, to dairy products, wines, beers, soft drinks, frozen food, and chocolate. Flake ice is used for the cooling of fresh fish.

9.1.1 Calculation method- RIEP

The methodology is based on food products. This choice has been made since FAO updates the database every year for food products produced and sold in each country. This data is needed for the "Inventories of the worldwide fleets of refrigerating and air conditioning equipment in order to determine refrigerant emissions".

For the Californian inventory, chocolate processes are not taken into account due to lack of information, but their contribution to the total bank and emissions is not significant. The only available information found was for the US is the following [CHOCUSA]:

Shipments (*million lbs)*	
1994	2.859
1995	3.02
1996	3.1

The consumption (*Consumption = Shipments + Imports – Exports*) [CHOCUSA] is also given for another three years.

Food domains taken into account are those of major importance: meat, dairy products, wines, beers, fishes, and frozen food. Cold storage is taken into account via two different routes:
- at the process facilities using a ratio between the cooling process and the properties of the product; it is therefore integrated in the cooling capacity dedicated to products,
- for general cold storage purposes, where needs are calculated separately.

Vegetables and fruits are taken into account in the cold storage and warehouse calculations. This choice has been made due to the very large difference that exists between crops and refrigerated vegetables and fruits. A calculation performed for the storage volume avoids large overestimates.

The global methodology (see Figure 9.1) used to determine the refrigerant inventories and emissions is based on data available for the production of all types of refrigerated and frozen food. The different food products are cooled or frozen at production sites, transported in refrigerated transport means, and then possibly stored in general warehouses. So, the food production data is used to establish the refrigerating equipment installed in the food industries.

The calculation steps are as follows.
- Analysis of the usual process design of a slaughterhouse, dairy, brewery, etc… to determine the installed refrigerating capacity.
- Definition of typical ratios of refrigerant charge referenced to the refrigerating capacity and the temperature level.
- Definition of the type of refrigerants selected which selection depends on the temperature level and on the type of country.
- Calculation of the refrigerant bank.
- Calculation of the refrigerant demand for new equipment (based on the equipment lifetime and the bank).

- Determination of national or regional emission factors applicable to the bank, yielding emissions.

9.1.2 Calculation of the cooling capacity and the refrigerant charge

Calculations are performed for the seven following sub-domains:
1. meat industry
2. dairy industry
3. wine and beers
4. flake ice for fresh fish
5. frozen food
6. warehouses.
7. soft drinks

Annex 4 presents detailed calculations for each sub-domain where the ratio between the cooling capacity and the annual production of a given product (kW/kg) is defined. Figure 9.2 summarizes the methodology and describes the relation between the annual production of refrigerated and frozen food and the refrigerant bank for all sub-domains.
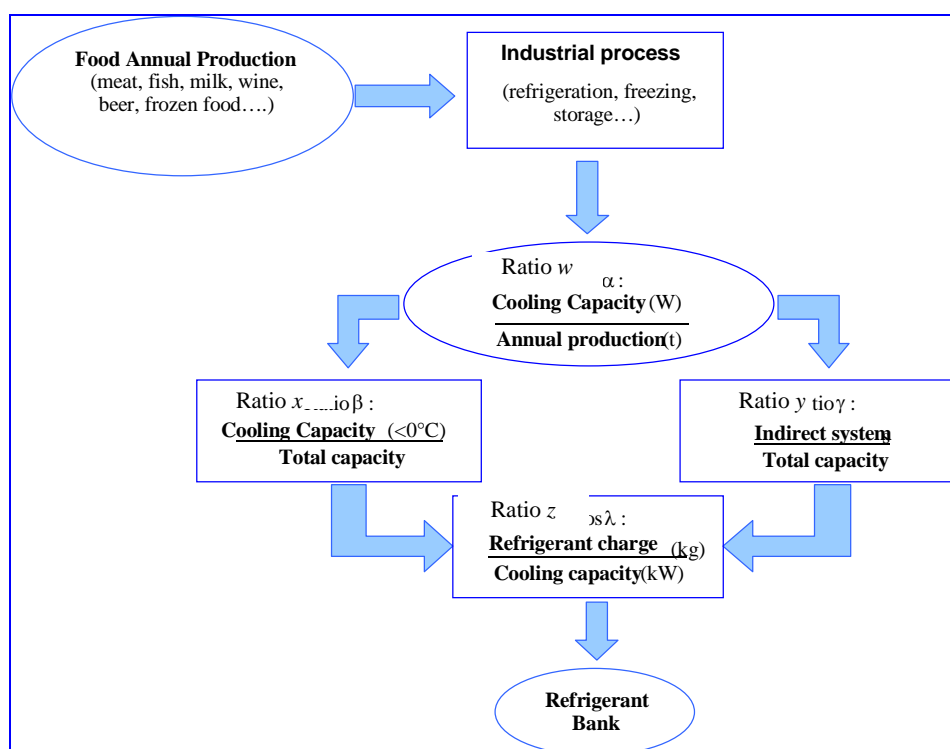

Figure 9.2 - Methodology for each type of refrigerated/frozen food industry.

The detailed studies of the processes applied in each sub-domain (see Annex 4) make it possible to determine the ratio between the total cooling capacity and the annual mass production of each sub-domain:

$$w = \text{Cooling Capacity (W) / Annual production (t)} \qquad (9.1)$$

Taking into account the division the cooling capacities for low and medium temperatures, which are again dependent on the cooling process, for each sub-domain the ratio x is defined:

$$x = \text{Low Temp. Cooling Capacity } (<0°C) \text{ / Total cooling capacity} \qquad (9.2)$$

Depending on the sub-domain and the technology, the indirect systems form a different part in the total. *y* is defined as the ratio of the cooling capacity of indirect systems and the total cooling capacity (direct + indirect):

$$y = \text{Cooling Capacity of indirect systems / Total capacity} \qquad (9.3)$$

The ratio *z* refers the refrigerant charge to the cooling capacity while considering the temperature level, and the technology (indirect or not).

$$z = \text{Refrigerant charge (kg) / Cooling capacity (kW)} \qquad (9.4)$$

The cooling capacity per mass of product and by level of temperature

Ratios defined in equations (9.1) and (9.2) are presented in Table 9.1 (see also Annex 4 for a justification).

Table 9. 1: Ratios *w* and *x* for the different sub-domains

	W **Cooling Capacity / unit**	**x** **w T Cooling Capacity / Total cooling capacity**
Meat industry	43 W/t	0.3
Dairy industry	12.9 W/t	0.2
Wine and beers	20.5 W/t	0
Flake ice for fresh fish	11.9 W/t	1
Soft drinks	4 W/t	0
Frozen food	35.8 W/t	1
Warehouses	33 W/m^3	0.7

The freezing capacity for meat is included in the calculations for the amount of frozen products globally. The cooling capacity in the meat industry is only defined for the production of fresh meat.

The Cooling capacity of indirect systems

The values of the *y* ratio as defined in equation (9.3) are given in Table 9.2 for each sub-domain, for the year 2004. This ratio is year dependent.

Table 9. 2: Ratio of indirect systems in new equipment in 2004

Industry	*y = Cooling Capacity of indirect systems / Total capacity*
Meat industry	0.15
Dairy industry	0.3
Wine and beers	0.15
Flake ice for fresh fish	0
Soft drinks	1
Frozen food	0.25
Warehouses	0.15

The refrigerant charge

The values of the *z* ratio as defined in equation (9.4) are given in Table 9.3. This ratio is given for medium and low temperature, for direct and indirect systems. Because of a lower liquid density, these ratios have to be divided by a factor 2 for ammonia.

Table 9. 3: Refrigerant charge referred to the cooling capacity

System	z = Refrigerant charge (kg) / Cooling capacity (kW)
Med Temp. Direct system	5.5
Low Temp. Direct system	8.8
Med Temp. Indirect system	1
Low Temp. Indirect system	1.5

9.1.3 Type of refrigerants

R-717 is widely used in the domain of industrial refrigeration (50% to 60% in US).
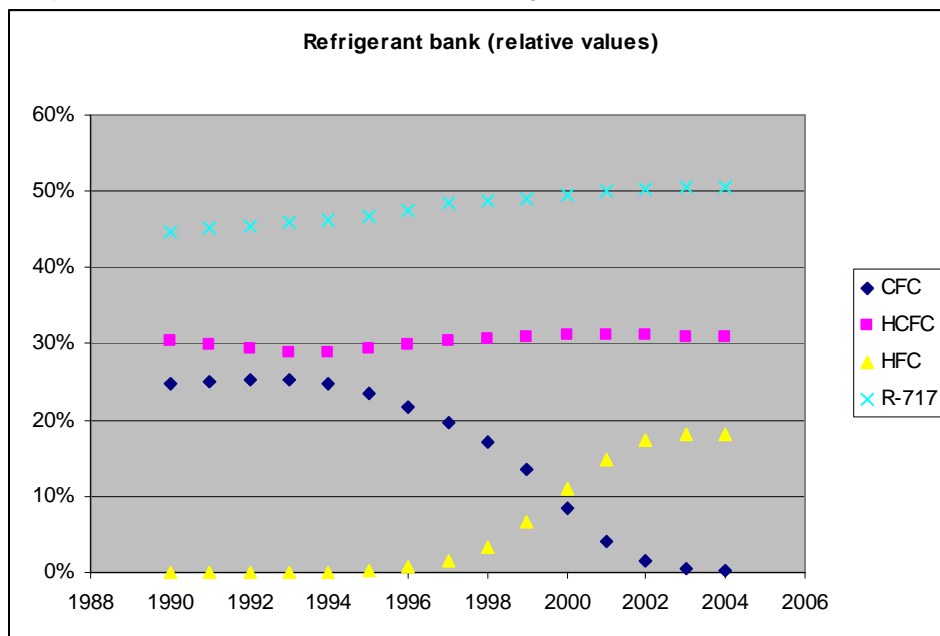

Figure 9. 1: Evolution of refrigerant bank in industrial refrigeration

9.1.4 Other characteristics

The average lifetime is used to establish the law of end-of-life of equipment.

Table 9. 4: Complementary data necessary to perform the RIEP calculations

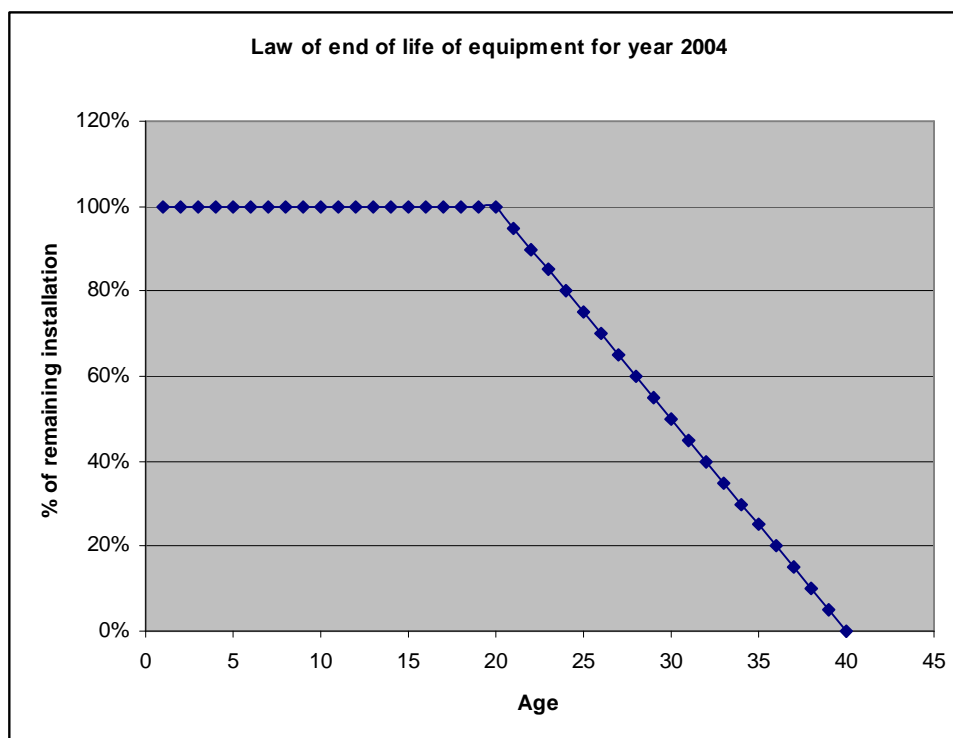| Year 2004	California
average equipment lifetime	30 years
annual fugitive emission rate	10%
Recovery efficiency	70%
percentage of charge emitted before servicing	30%

Figure 9. 2: law of end of life of refrigeration equipment

9.1.5 Production data for food products

Production data is available for the following sub-domains

Meat industry

> Data on production of meat is available for the USA from the FAO [FAO01] and USDA [USDA01]. For California, such data is available from the USDA for many years, except for chicken meat. A compilation of both sources allowed establishing the total meat production for California.

Table 9. 5:meat production in US and California

Meat production (t)	1990	2004
USA	39,747,784	49,966,545
California	2,058,102	2,637,806

Dairy industry

The dairy industry covers the production of milk, butter, cheese and cream.

California is a leading state in milk production, covering approximately 20% of the total nation milk production.

Both USDA [USDA02] and FAO [FAO01] sources were used in order to compute the total industry dairy production.

Historical data for US are available. For those unavailable values for the state of California, ratios for cheese, butter and cream production over the USA production were established (eg. Cheese Production US (t) /Milk Production US(t)) were established, and applied to the milk production in California in order to compute the cheese and other unavailable dairy products historical statistics for California.

Table 9. 6: dairy production

Food Production		1990	2004
Milk production (t)	USA	67,005,118	77,534,358
	California	9,501,399	16,540,246
Cheese production (t)	USA	2,748,516	4,024,793
	California	317,054	903,928

Wine and beers

California produces more than 90 percent of total US wine production [WINST]. Production data is available for California from the Wine Institute [WINST] for years 1986 till 2004. The FAO provides statistics for the USA starting from the year 1961. Both sources were used to compute the historical production for California.

Table 9. 7: wine production

Wine production (t)	1995	2004
USA	1,654,354	2,304,817
California	1,502,967	2,070,724

Information on beer production is available for the USA from the FAO [FAO] and the Beer Institute [BEINST]. The total USA production of beer is about 23 Millions of metric tons. The Beer Institute gives data for California for many years. In 2004, the California beer production accounts for about 11% of the global US beer production. This ratio is almost constant for years 1999 till 2006, and was therefore applied to estimate historical values for California.

Flake ice for fresh fish

In the calculations, flake ice production is directly linked to the daily catch of sea and river fish. NOAA's National Marine Fisheries Service provides statistics on domestic commercial landings [NMFS] for the USA and California, and the FAO gives global production statistics.
Refering to the NOAA's data, the fish landing in California accounted for 31% of the total US landings in the year 1950. In the year 2005, the share of California in the total US landings decreased to reach 5%, while the state of Alaska became the most important producer with 60% of the total US landings. Both sources were therefore used to calculate the production of California.

Table 9. 8: fish production

Fish production (t)	1990	2004
USA	6,096,539	435,438
California	5,972,841	233,633

Frozen food

The FAO provides the quantity of frozen food produced in the USA, i.e. 13.8 Millions of Metric tons for the year 1990 and 16.3 for the year 2004. Based on the US Bureau Census, there are about 14% of the USA frozen food industries in California. Due to lack of information regarding the quantity produced in California, this ratio was adopted.

Warehouses.

The volume of refrigerated warehouses is available from the USDA for the USA and for California for many years. Linear interpolation is done for the years that are not given by the USDA.

Table 9. 9: Warehouse capacity

Net refrigerated warehouses capacity (m^3)	1990	2004
USA	55,578,891	73,086,802
California	6,015,404	9,137,563

Soft drinks

The US Bureau Census gives the total number of establishments engaged in manufacturing soft drinks and artificially carbonated waters for the USA and for California. The ratio *Industries in California/Industries in USA* = 11% is applied to the total volume of CSD US industry given by the Beverage Disgest [BVDIG].

Table 9.10: Carbonated soft drinks production

	CSD volume (192 billion oz cases)		CSD volume (t)	
	1990	2004	1990	2004
USA	6.1308	7.84384	33,371,171	41,619,252
California	0.674388	0.8628224	3,670,829	4,578,118

9.1.6 Milk tanks

Milk tanks are installed on farms and represent a sub-domain included in the dairy domain. The calculations are specific, based on the number of milkings, which enables to define the storage volume of the milk tank. Knowing the storage volume, it is possible to define the refrigerating capacity and the refrigerant charge; the method and additional data are presented in Annex 4.

Average refrigerant charge

The average charge of refrigerant is 2.09 kg/m^3 of storage.

Characteristics

Table 9. 11: Characteristics of milk tanks

Average Lifetime (years)	Annual Emissions (%)	Recovery Efficiency (%)	Charge Emitted before Servicing (%)
15	5	50	30

Refrigerant type

Table 9. 12: refrigerant distribution on the market

Year	% of R-12 in Market	% of R-404A in Market	% of R-22 in Market
1990	20	0	80
1991	10	0	90
1992	0	0	100
1993	0	0	100
1994	0	0	100
1995	0	0	100
1996	0	0	100
1997	0	0	100
1998	0	0	100
1999	0	10	90
2000	0	20	80
2001	0	20	80
2002	0	20	80
2003	0	20	80
2004	0	20	80

9.2   Industrial processes (other than food industry)

9.2.1 Data sources and detailed calculations

Refrigerating needs in industrial processes other than food processing are multiple. They cover a broad range of temperatures. Two types or categories of refrigerating equipment have been defined and analyzed.
1. Chillers operating at a temperature above 0°C. A large amount of this equipment is bought "from the shelf", and one only needs to define the capacity and the level of temperature. These chillers cover 55% of the refrigerating capacity needs.
2. Refrigerating systems particularly designed for low temperature applications, where the process specifications are well taken into account.

In order to avoid double counting, chillers are not taken into account in this section because the demanding data for the chiller production and the chiller demand normally merge all different chillers types, i.e., the ones for comfort cooling and the ones for industrial processes (see Section 8.2).

In this section only the low temperature refrigerating systems installed in the chemical industry are considered for the following reasons:
▪ it is a more significant domain (except food) for low temperature applications,
▪ the important industrial domains such as tire manufacturing, electronics, etc. use chillers only to cover their cooling needs.

A thorough analysis of the installed base of a chemical companies (under confidentiality agreement) has enabled the development of a typical scheme of an industrial production site. Based on this study, the low temperature cooling capacity has been projected to all other chemical manufacturers, in order to have a first estimate. The characteristics are presented in Table 6.12.

Table 9.13: Refrigerant charge and cooling capacity for a chemical plant

	Medium temperature	Low temperature
Cooling capacity	55%	45%
Ratio (kg/kW)	2.3	5.5
Refrigerant charge	40%	60%

Even if many installations may have a lifetime of more than 30 years -taking into account the big overhauls-- the lifetime of equipment is considered to be 15 years, i.e., the time before a significant maintenance takes place.

Table 9. 14: Other characteristics of typical refrigerating systems installed in chemical plants

Life (years) Before remodeling	Annual Emissions (%)	Recovery Efficiency (%)	Charge Emitted before Servicing (%)
15	10	50	30

Based on available information collected from the websites of the main chemical companies, operating globally, the French inventory for chemical industries was developed. The Californian inventory for this sector was then derived from the French one by applying GDP ratios.

Table 9.15 describes the evolution of refrigerants in use for the new refrigerating systems installed in the chemical industry.

Table 9. 15 refrigerant distribution in industrial processes

Year	% of R-404A	% of R-11	% of R-12	% of R-22	% of R-717
1990	0	4	34	60	2
1991	0	4	34	60	2
1992	0	4	34	60	2
1993	0	3	34	61	2
1994	0	1	25	72	2
1995	0	0	0	98	2
1996	0	0	0	98	2
1997	0	0	0	98	2
1998	0	0	0	98	2
1999	0	0	0	98	2
2000	0	0	0	95	5
2001	0	0	0	95	5
2002	5	0	0	90	5
2003	10	0	0	85	5
2004	10	0	0	85	5

9.2.2 Results of calculations: refrigerant banks, and emissions
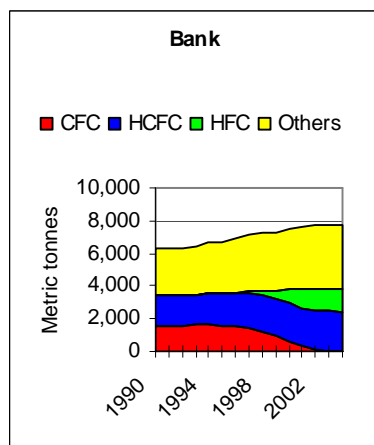
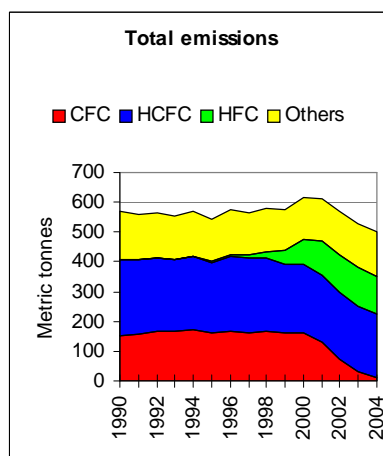Figure 9. 3: refrigerant bank in industry
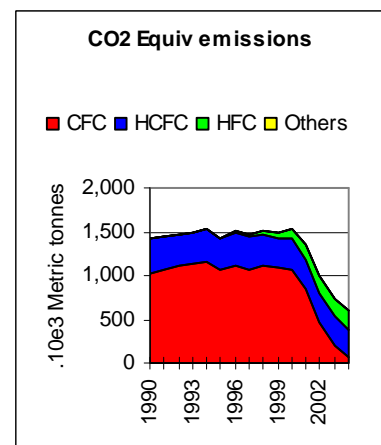

Figure 9. 4: refrigerant emissions in industry

Figure 9. 5: CO$_2$ equivalent emissions in industry

Ammonia is widely used in industrial refrigeration. Bank of ammonia approaches 4,000 metric tonnes in 2004. Emissions of HCFC totalizes near 250 metric tonnes in 2004. The phase out of CFCs, when started in 1996 with the retrofit of existing systems have a high impact on CO$_2$ equivalent emissions.

9.3   Overall results

The Mobile air conditioning sector as well as the refrigerated transport sectors that are out of the scope of the present report have been established nevertheless in order to have an overall picture of refrigerant inventories in California. Moreover, the inventories have also been done at the US level in order to compared the refrigerant demands per refrigerant as derived from this work and the refrigerant sales per refrigerant as possibly known by refrigerant manufacturers. In the next weeks a comparison will be performed with confidential data gathered from refrigerant manufacturers.

9.3.1 Refrigerant bank

The refrigerant bank is presented here by refrigerant types. The type "others" is ammonia mainly used in the food industry.

- Figure 9.6 shows the evolution of the refrigerant bank per type of refrigerants. It is clearly seen that the refrigerant bank follows the same growth as the population and GDP shown in figures 7.1 and 7.2.

- In 2004, the refrigerant bank in California is estimated at 150,000 metric tones and it was about 80,000 tones in 1990.

- CFCs are not in use anymore, insignificant contribution in 2004, but R-22 and HFCs have been continuously since 1995.

- The dominant refrigerant in the overall bank is R-22 which constitutes 75% of the bank.
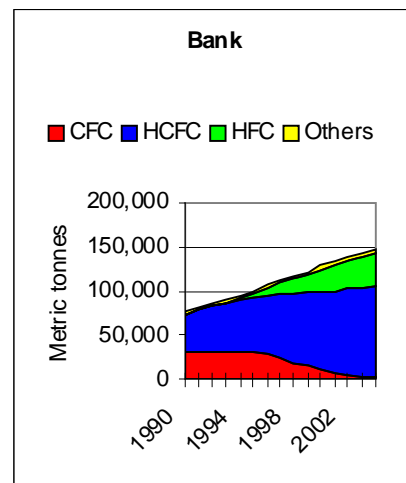

Figure 9.6 California refrigerant bank per refrigerant types

When analyzing the repartition of refrigerants in the different sectors, stationary air conditioning constitutes clearly the dominant sector with nearly 100,000 tones and so 75% of the total bank of refrigerants. Within this sector roof-tops constitute the dominant part of air to air systems.
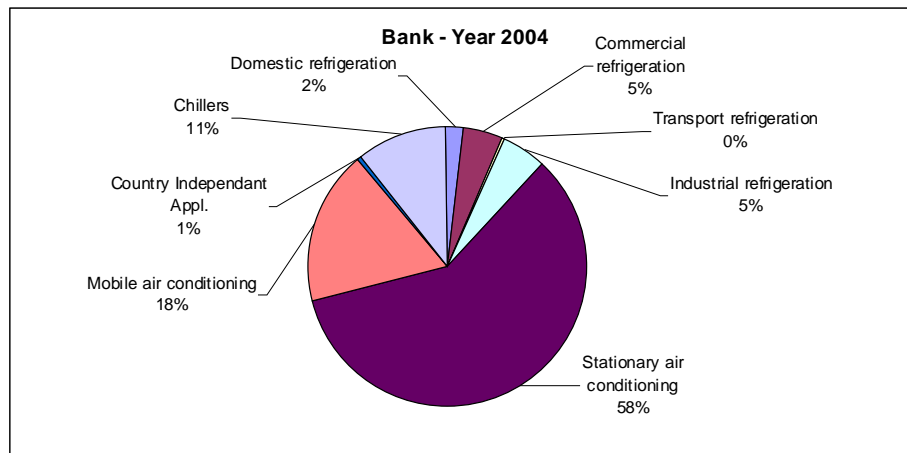
Bank - Year 2004

Domestic refrigeration 2%
Chillers 11%
Country Independant Appl. 1%
Mobile air conditioning 18%
Commercial refrigeration 5%
Transport refrigeration 0%
Industrial refrigeration 5%
Stationary air conditioning 58%

Figure 9.7 Structure of the refrigerant bank per sectors

Note : the "country independent application" corresponds to refrigerated containers which are coming in California, which are serviced but international refrigerated containers are difficult to attribute to countries.

9.3.2 Refrigerant emissions

Refrigerant emissions are defined for each sector and more precisely for each type of equipment as presented in section 7 and in the annex 2.

- From 1990 to 1997 CFCs represent 50% of the emissions, and the other 50% being mainly HCFCs

- From 2000, the emissions are decreasing based on the assumption of recovery at end of life (see section 9.3.4) and to the fact that new equipment requires less servicing during the first years of use. A mature servicing market with new refrigerant requires at least 15 years of operation.

- The replacement of CFCs and of some HCFCs by HFCs and a better initial containment explain the decreasing trend.

Total emissions

■ CFC ■ HCFC ■ HFC □ Others

Metric tonnes

30,000
25,000
20,000
15,000
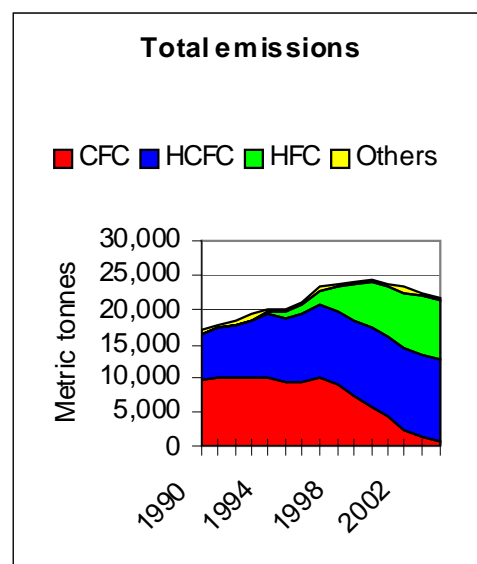10,000
5,000
0

1990 1994 1998 2002

Figure 9.8 Refrigerant emissions per refrigerant types

Figure 9.10 shows the impact of different emission factors depending on the application sector : Mobile AC which represents 18% of the bank contributes to 38% of the overall emissions. This sector been more emissive, thanks to the use of small cans for servicing and of emissive components as the shaft seal of the open type compressor.

On the contrary Stationary AC systems whose emission factors are lower, represent 58% of the overall bank, and only 38% of the overall emissions. The main progresses have to be made for the recovery at end of life, even if lessons learnt from the field are necessary to understand what are the components to be improved in term of leak tightness.
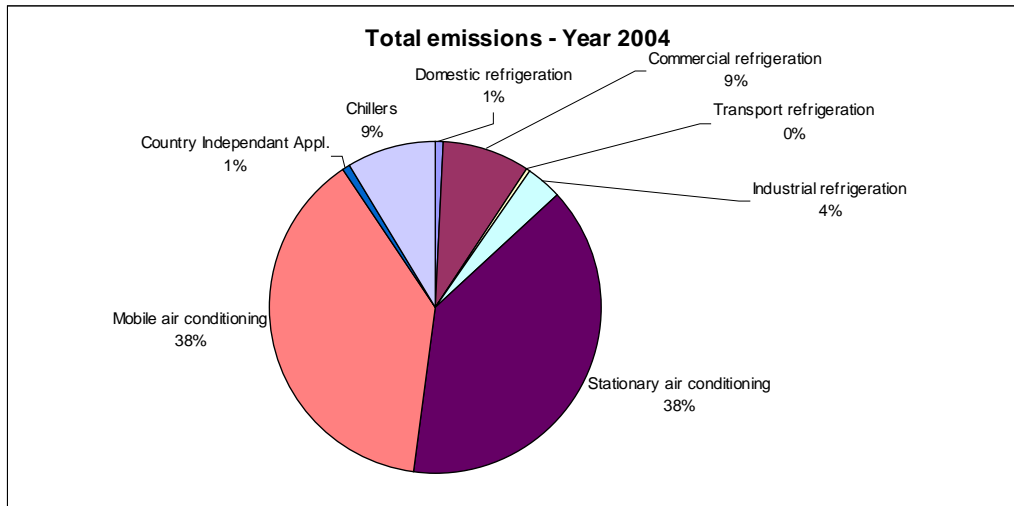

Figure 9.10 Refrigerant emissions per application sectors

9.3.3 CO_2 Equivalent emissions

When looking at figure 9.11, the paramount factor is the change from CFCs having high GWPs and specially R-12 (10,600) to HFCs mainly R-134a (1,410) resulting in a dramatic decrease of Equivalent CO_2 emissions. It is obvious that the absence of accounting of CFCs and HCFCs in the climate convention is not physically justified.

- Until 2002, the CO_2 equivalent emissions are dominated by R-12 with its high GWP and it is mainly associated with mobile air conditioning systems, domestic refrigeration and small commercial equipment.

- Note : a systematic and efficient recovery policy at end of life of equipment could have limited a so significant effect on climate change.

- In 1997, emissions reach a maximum at nearly 100 million metric tones CO_2 equivalent.

- In 2004, emissions drop down to 33 million metric tones CO_2. Equivalent.
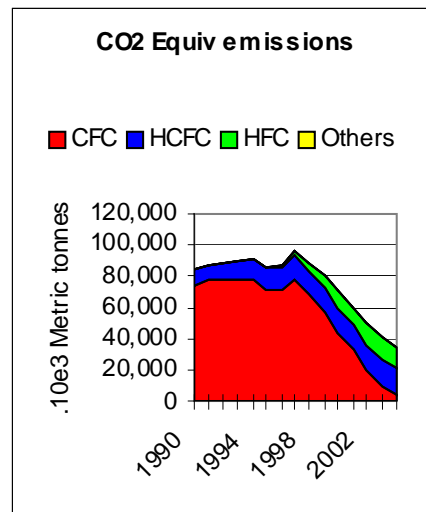
CO2 Equiv emissions

CFC ■ HCFC ■ HFC □ Others

Figure 9.11 Refrigerant emissions expressed in CO_2 equivalent

- MAC sector is the first domain with 41% of CO_2 equivalent emissions in 2004, then comes stationary AC with 36% of global emissions.
- Commercial refrigeration is 5% of the refrigerant bank but 10% of CO_2 equivalent emissions. It has to be noted that if R-404A (GWP = 3,780) is going to replace systematically R-22 (GWP = 1,700) the increase in CO_2 equivalent will twice the one of R-22 in the next 10 years.
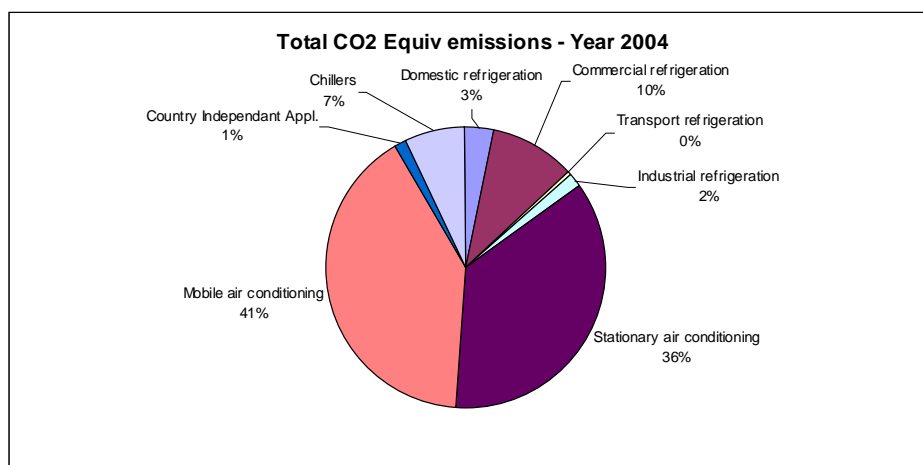
Figure 9.12 Refrigerant emissions per sectors and expressed in CO_2 equivalent

When comparing figures 9.10 and 9.12, due to the nearly nil contribution of CFCs in 2004 in the refrigerant emissions, the repartition of emissions expressed in CO_2 equivalent is not that different compared to the emissions expressed in refrigerant tones.

9.3.4 Refrigerant recovery

Figure 9.13 requires that verifications based on distributors and reclaimers data have to be carried out in order to verify the assumptions taken in this report.

- Refrigerant recovery is considered to be effective since 1996, with CFCs phase out.

- Refrigerant recovery is estimated to be around 3,000 metric tones in California in 2004

- These quantities include refrigerant recovered and recycled on site and refrigerant recovered and regenerated at the manufacturer plant. It is acknowledged that the need of CFCs and HCFCs have been and are motivations to recover for reuse and keep the old refrigerating systems in operation with the refrigerants no more available on the market.
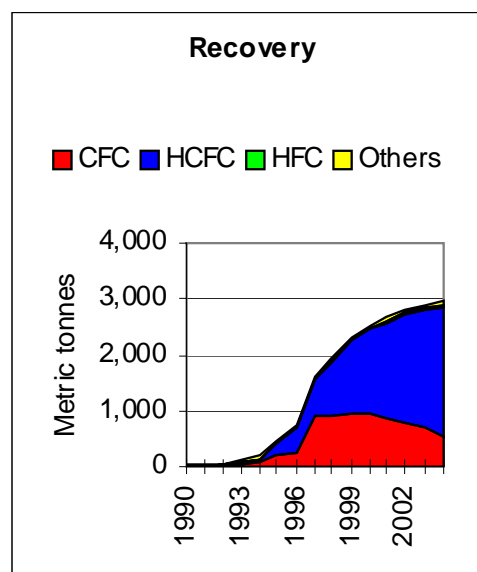

Figure 9.13 Recovery per refrigerant type